Design & Development of Expert System Based Framework for Servo Actuator Mechanism

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ISLAMABAD

SEPTEMBER, 2015

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A thesis submitted in partial fulfillment of the requirements for the degree of

MS Mechanical Engineering

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Declaration

I certify that this research work titled "*Design & Development of Expert System Based Framework for Servo Actuator Mechanism"* is my own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources it has been properly acknowledged / referred.

Signature of Student

Laeeq-ur-Rehman

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Language Correctness Certificate

This thesis has been read by an English expert and is free of typing, syntax, semantic, grammatical and spelling mistakes. Thesis is also according to the format given by the university.

Signature of Student

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Signature of Supervisor

Copyright Statement

Acknowledgements

I am thankful to my Creator Allah Subhana-Watala to have guided me throughout this work at every step and for every new thought which You setup in my mind to improve it. Indeed I could have done nothing without Your priceless help and guidance. Whosoever helped me throughout the course of my thesis, whether my parents or any other individual was Your will, so indeed none be worthy of praise but You.

I am profusely thankful to my beloved parents who raised me when I was not capable of walking and continued to support me throughout in every department of my life.

I would also like to express special thanks to my supervisor Dr. Hassan Aftab Saeed for his help throughout my thesis and also for Finite Element Methods which he has taught me. I can safely say that I haven't learned any other engineering subject in such depth than the ones which he has taught.

I would also like to pay special thanks to Dr. Wasim Akram Tarar for his tremendous support and cooperation. Each time I got stuck in something, he came up with the solution. Without his help I wouldn’t have been able to complete my thesis. I appreciate his patience and guidance throughout the whole thesis.

I would also like to thank Col Dr.Aamer Baqai and Mr. Raja Aamer Azim for being on my thesis guidance and evaluation committee. I am also thankful to my colleague Mr.Sufian Hijaz for his academic, moral and logistic support.

Finally, I would like to express my gratitude to all the individuals who have rendered valuable assistance to my study.

Dedicated to my family especially my dearest Mother, my honorable father Mr.Lutfur Rehman & adored siblings and children (Shahzanin and Hurain) whose tremendous support, love and cooperation led me to this wonderful accomplishment.

ABSTRACT

Design and development of the zero backlash servo actuator mechanism and to develop an expert system which will evaluate the data analysis of the servo actuator mechanism which will evaluate the results with finite element analysis & experiments which will optimize the design calculations. Design of 6 DoF platform which has high payload up to 3 ton capacity and can have speed of 0.381 m/s for minimum ball screw length of 0.75 meter. Develop an expert system which evaluates the results of coded expert system by comparing it with the results of finite element simulations and to deliver an optimum design for specific requirements.

Key Words: *Ball screw, Expert System, Finite Element Analysis, Experimental Testing of Buckling*

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Chapter 1 : Introduction

1.1 Six Degrees of Freedom Platform

The 6 DoF Stewart platforms have been developed in 1954. In the year 1965 for flight simulator motion in 6 DoF there was used mechanism called parallel prototype. The mechanism was driven by two means electrically or hydro-electrically. Although, Stewart platform has been used very often in the application and most of researcher used Stewart platform for kinematic and dynamic studies of the 6 DoF motion and orientation.

In hydraulic application there is a great propel with the combination of the hydraulic and control mechanism. This is because that hydraulic are more favored than the electro mechanical driven machineries in the automobile applications. Hydraulic can produce large forces with very high speed and response is very rapid too in this case.

For compensating non-linearity in the system control techniques are used in electro-hydraulic servo-systems. Researchers anticipated the Nonlinear control techniques for hydraulic servo-systems employing linearization..

Most of the preceding efforts related to pressure and force control has been used in the electro hydraulic actuator of servo mechanism and has been implemented using Lyapunov analysis. Hydraulically actuated manipulator which is modal based controller has been studied, but desired position was calculated rather than the actual in the feed forward controller. Preceding work on a model based motion and force controller analysis have good response of tracking.

Because there are different nomenclatures and definitions used by different researchers in the field of parallel mechanisms, it is quite necessary to define the nomenclature used in current research. The following terms are defined:

Parallel Mechanism*:* where End-effecter is connected to the base by minimum two independent kinematic chains It is called closed-loop mechanism and it is also called Parallel Platform or a Parallel Kinematic Mechanism (PKM).

Base Platform*:* The immovable plate of the parallel platform; it is also called the fixed platform.

Top Platform*:* The moving body connected to the base platform via extensible legs. It is also called moving platform or mobile platform.

Legs: The independent kinematic chains in parallel connecting the top platform and the base platform also called a connector.

Joint: Kinematic connection between two rigid bodies providing relative motion.

Examples of High Speed Hexapods

Mistral High Speed Hexapod

Mistral is high speed hexapod which has maximum 1 ton Payload capacity, angular travel of 30 degree, linear travel up to 470 mm with the speed of 1 m/s, acceleration of 1 m/s2and accuracy of 0.5 mm.

Figure 1.1:MISTRAL Hexapod

Sirocco Hexapod

Mistral is high speed hexapod which has maximum 2 ton Payload capacity, angular travel of 30 degree, linear travel up to 470 mm with the speed of 2.5 m/s acceleration of 6 m/s^2 and accuracy of 2 mm.

Figure 1.2: Sirocco Hexapod

1.2 Ball Screw System

Ball screw has high efficiency and has a long life and that is why it is used in the precise motion of object following 6 DoF. In the high speed ball screw heat is generated from contact points of ball screw and ball nut due to which thermal expansion occurs. Accuracy of system decreased with heat generation.

The ball screw research was outcome of the increasing necessity of precise motion controls in engineering application. Reciprocating ball screw mechanism is device for motion and force transfer and is very advantageous for the mechanism for achieving high positional accuracy at high speed of the shafts. Therefore it can be used in the feed drive mechanism of tools and work piece in manufacturing industry and precise degree measurement of the system required precise alignment. The high speed of translation of the ball screw shaft is of main importance in the high speed processing device of the motion and transfer mechanism. The ball achieve linear velocity greater than 100 m/min and acceleration greater than the 9.8 m/s2.for this situation high speed ball screw should be designed with preload for high precise motion control and positional accuracy. With increase in preload the friction between ball screw elements increases which produce large heat generation with the requirement for high torque. Therefore, a vital design criterion must be developed to decrease slip of balls in the ball screw during high speed of ball screw with minimum heat generation with requirement of small driving torque to maximize the efficiency of ball screw mechanism.

Ball screw consists of three basic components - shaft, nut and balls in thread between them shown in Figure 1.3.The ball screw is widely used in machine tool and other industries. The most important advantage of ball screw is high positioning accuracy

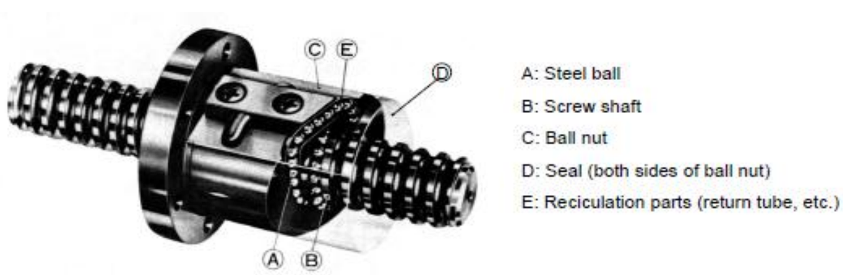


Figure 1.3: Ball screw parts detail

1.2.1 Screws

In machine design, the screws can be treated as block sliding on wedge or an inclined plane. There are two angles of importance:

The Lead Angle: The lead angle can be calculated by the inverse tangent of L, the Lead, or distance the screw moves with one rotation of ball screw divided by the circumference of the ball screw cross section. This angle is usually designated as alpha α.

The Friction Angle: The friction angle can be calculated by the inverse tangent of the coefficient of friction μ. This angle is usually designated as ϕ.

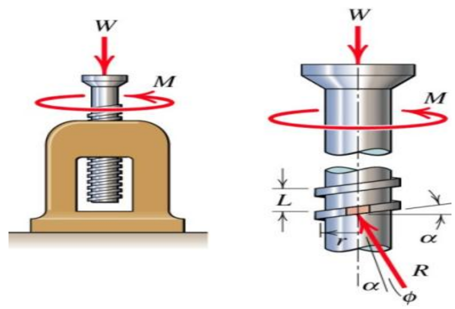


Figure 1.4 : Lead angle (α) and Friction angle(ϕ)

The resultant force *R*, acts on the thread at a total angle of $(\alpha + \phi)$.The moment of R about the axis of the screw is $R.r\sin(\alpha + \phi)$.We must consider the total force exerted by all of the threads $\sum R$. Hence, the total moment due to all reactions of the screw thread becomes $\sum R r \sin(\alpha + \phi)$.

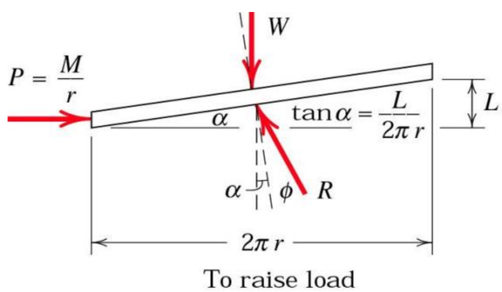


Figure 1.5: Verification of the Kinematics for the Stewart Platform

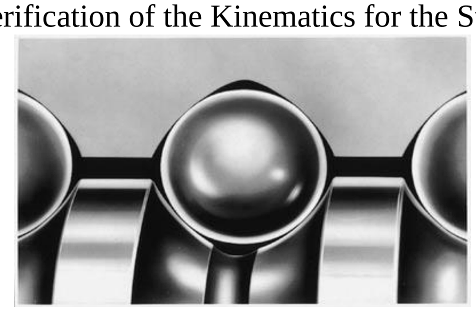


Figure 1.6: Balls between Nut & Screw

1.2.2 Screw Types

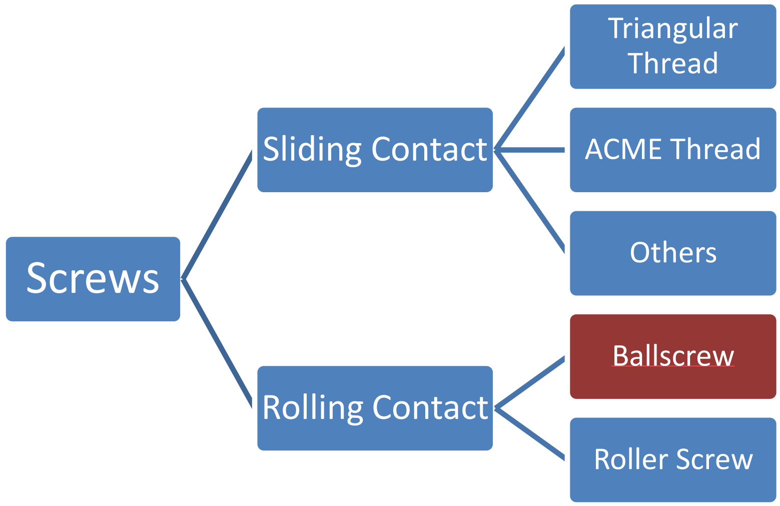


Figure 1.7: Type of Ball Screws

1.2.3 Ball Screws

Ball screws are in the family of power-transmission screws in which there is transfer of force and motion-between the ball screw mechanism. The operation methodology is same as the conventional power screws but in these screw types the sliding friction has been replaced by the rolling friction.

Figure 1.8*:* Ball Screw

These type of screws have grooves acting as the inner race. The nut has internal grooves that act as the outer race. One part between ball screw or ball nut is fixed and the other part rotates. This action changes the torque to thrust. Ball-screws can be classified on the basis of three parameters:

Axial Screw: The degree to which a ball nut can be moved in the screw axis without rotation of either nut or the screw.

Preload: Process of increasing stiffness of the ball screw to remove backlash.

Load Accuracy*:* The degree to which rotational movement is transferred into linear motion.

1.2.4 Advantages of Ball Screws

- Ball screw actuators are more efficient than lead screws.
- Ball screw need lower power requirements as it uses smaller motors, clutches, and gears.
- Ball screw actuator has low cost than hydraulic-pneumatic actuator. Ball screw work more silently and has no requirement for pumps, hoses, fluids, or pneumatic air as work without any fluid power.
- Cables, belts and chain system are affordable and less cost than ball screws, when belt stretches due to wear, they are also less precise which leads to inaccurate degree measurement and positioning.

1.2.5 Description detail Ball Screws

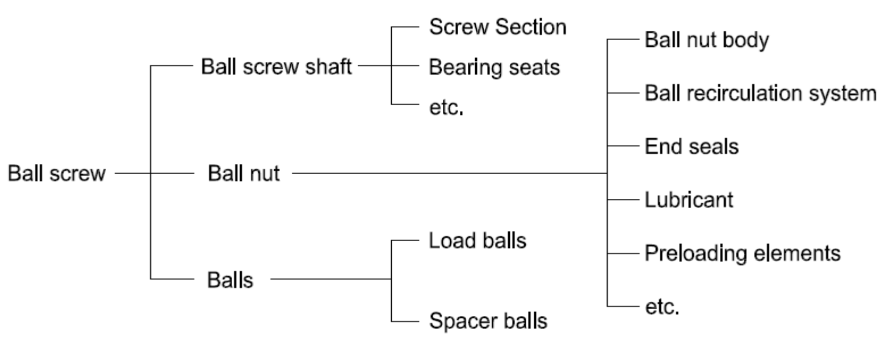


Figure 1.9 : Description of Ball Screw parts

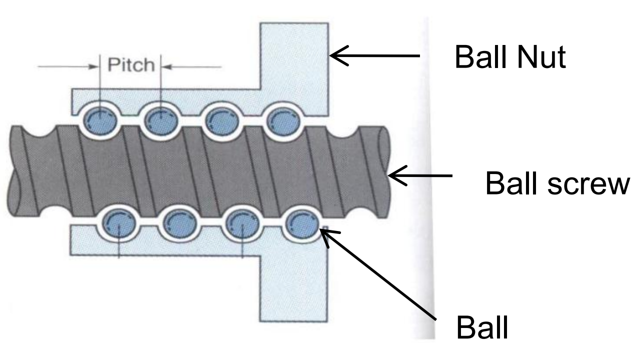


Figure 1.10*:* Ball screw cross section view

1.2.6 Ball Screw Free Body Diagram

Ball screw free body diagram with ball nut and thrust bearing has been shown below in Fig.

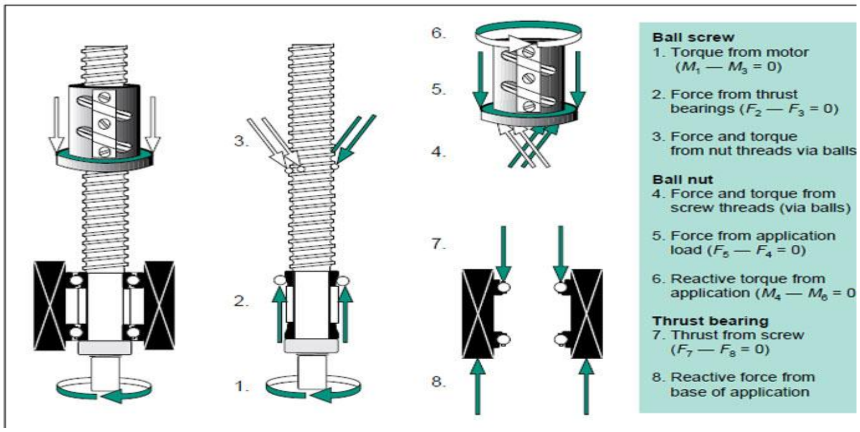


Figure 1.11: Ball Screw Free Body Diagram

1.2.7 Applications of Ball Screws

Ball screws are used in a various engineering applications. Ball screw can be used in high and low load application due to its high mechanical efficiency. They are used in mechanisms such as flight simulators, antenna precision alignment, in propellers for controlling pitch, wings flapping using a single ball screw mechanism, nose landing of aircrafts and main landing gears etc. Aerospace technology use ball screw due to its high precision, high mechanical efficiency low friction and low heat generation. Although ball screw mechanism has low demands in automobile and construction as of low efficiency industrial applications. In industries ball-screw is using in CNC milling and lathe machine tables, robotics, and transport mechanism used for automation. Miniature ball screws are extensively used in the automobile the size of which is small as 0.375 in and lead of ball screw as small as 0.125 in with 90 % efficiency. Ball screw is used in various fields and various types of applications.

- Aeronautics
- Automotive
- Defense Industry
- Naval industry
- Nuclear
- Spatial (Satellite dish positioning)

1.2.8 Zero Backlash Ball-screws

A zero-backlash ball screw is a long, threaded device that rotates to provide precise linear motion control with minimum friction. Low friction in ball screws yields high mechanical efficiency. A zero backlash ball screw may be 100% efficient, versus 50% efficiency of an Acme lead screw of equal size. Lack of sliding friction between nut and screw especially in zero backlash systems, reduces downtime for maintenance, part replacement and decreases demand for lubrication.

1.3 Servo Motors

A servo motors are used in servomechanism which is extensively used for controlling the movements with information of feedback devices. There is large number of servomechanism and the type we use here will give us the translation motion with these ball screws and servo motor. . Ball-screw servo mechanisms are mainly used in every engineering fields such as CNC machine tools, automobile and aerospace technology. A ball-screw servo motors mechanism consist of ball-screws and ball-nuts, sensors for feedback information ,gears ,servo motor ,mechanical joints and motion controllers .

Mechanical subsystem and Control sub system are two main division of servo mechanism using ball screws. Mechanical system consists of the structure and mechanics for driving structure components. The 2nd is the control subsystem composed of sensors for feedback information and controllers for motion. Servomechanisms Performance is mainly dependent on interconnection between the two subsystems.

A type of motor used in applications that require precise positioning used for controlling movements of platform in 6 DOF High-torque, low-speed motor is used for high efficiency and accurate degree measurement of the platform motion

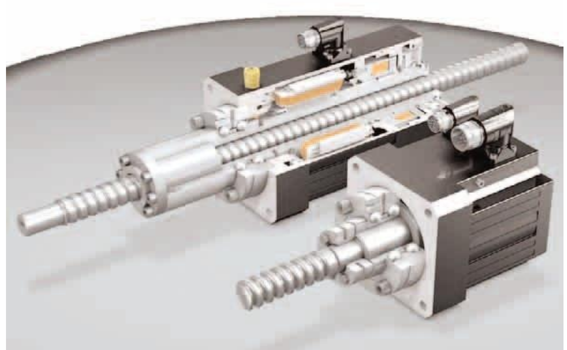


Figure 1.12: Integrated design of Servo Ball Screw

1.4 Expert System

Calculations often become very tedious and a cumbersome job. Intelligent expert system solves complex problems. Expert system does not evaluate the result of the system Specific information e.g. 6 DOF platform Errors may occur in the knowledge base (Coded Expert System) as the data for specific output is often an overdesign. Evaluation makes the expert system very useful for the industry. Increase decision quality, consistency and speed of decision making. In manufacturing industry, 6-DoF platforms are often used for motion and positioning of high payloads. With focus on the application in manufacturing environment, we seek to develop one such platform capable of moving payloads in all six degrees of freedom. However, design of such systems usually requires a considerable effort in terms of calculations, which often becomes a cumbersome job. In order to reduce the time and effort required for design of 6 DOF platforms, intelligent expert systems are developed that are able to solve complex problems. One drawback of such expert systems is that they are unable to evaluate the effectiveness of design results. Errors may occur in the knowledge base (Coded Expert System) as the data for specific output is often an overdesign.

Hence, we also aim to develop an Expert System which evaluates the results and optimizes the design calculations. Expert System (ES) is a knowledge base system that uses the expert's knowledge to help work men in executing various functions in various fields. ES accesses database in an intelligent manner. It can be examined for correctness, consistency and completeness. The knowledge after examination/analysis is adjusted, which improves the quality and evaluates the ES.

1.5 Motivation

6-DoF platforms are used for motion and positioning of objects following six degrees of freedom. They allow for high precision control over the movement and orientation of the objects being carried with respect to the surrounding environment. These platforms have applications in a wide variety of areas including medical technology, aircraft simulators, manufacturing industry, etc.

DOF platform used extensively in the Antenna precision alignment and motion control for telescopes, application in medical equipment for diagnosis/surgery, aircraft simulators for motion in six degrees of freedom. Load capacities double when using two nuts.

Ball screw can be used in numerous innovative chassis applications.

In electromechanical steering system ball screw offers greater comfort, lower cost, reduced fuel consumption. Ball screw is main key actuation element on the F-14 Tomcat for the variable sweep wings. A Ball screw is considered standard practice for actuating trailing edge flap systems, horizontal stabilizers and landing gear.

1.6 Problem Statement

To validate the results of FE simulations (stress analysis, modal analysis etc.), we shall resort to experimental structural testing. Results from experiments will serve as a standard for authentication of numerical simulations. Procedure validation to be done for Servo actuation testing ,Once the FE-based simulation results are verified, they will then be used as a standard for evaluation of the coded expert system, and for any adjustments/corrections that may be required in the expert system which can predict system for any combination of input data.

1.7 Objectives

The main objective of the research is to Design Six DoF platform which has high payload up to 3 ton capacity and can have speed of 3 m/s for minimum ball screw length of 0. 5 meters and to Develop expert system this will evaluate the results of coded expert system by comparing it with the results of finite element simulations and to deliver an optimum design for specific requirements.

Chapter 2 : Literature Review

2.1 Six Degrees of Freedom

The 6 DoF Stewart platform has been developed in 1954 [1], [2]. . In the year 1965 for flight simulator motion in 6 DoF there was used mechanism called parallel prototype [3]. Various numbers of studies of this Stewart mechanism and its differences have been presented in [4]. Mohammad Kasim Abdul Jalil [5] works out on prototype for motion platform used for vehicle driving. The vehicle is delivering different motion during driving on different road condition .the simulator is not used only for driving simulator but can also be used for the mini ships and aircrafts.

Driving simulator developed can also be used for shaking table during earthquakes, platform for vibration and seismic studies for research. Shaking table and vibration apparatus can be used as for shaking structures or modals of structures and items with simulated ground shaking. Ioannis Davliakos and Evangelos Papadopoulos [6] develop a tracking controller for position of six-degree-of-freedom (DoF) using inner force loop for tracking and based on fast modal electro-hydraulic Stewart platform.

Electro-hydraulic actuator mechanism using friction and servo motor properties and verified by integrated system calculations. Linearization approach for controlling is implemented for non-linear input and output of the mechanism. Leakage of hydraulic used in the system and friction between parts of mechanism has been taken into consideration.

Daniela Gewald [7] presented a strategy for combined tracking and vibration control of a parallel mechanism. The presented nonlinear feed forward feedback controller has shown in its high degree of performance for the combined stabilization and tracking of a hexapod system the investigated mechanism is hexapod system with six degrees of freedom that can realize accurate motions of the tool centre point (TCP) to compensate mechanical vibration transferred from the environment to the TCP an active stabilization scheme both feed forward –feedback controller is applied.

the feed forward controller should compensate most of the vibration forces and decouples the nonlinear hexapod dynamics ,whereas the feedback controller compensate residual TCP vibration and provide the desired tracking task. Sisamak pedrammehr, mehran mehboub khah, Navid khani [8] addressed analytical study on the vibrations of a parallel manipulator. Moving platform vibration calculations, stiffness and damping of the actuators are taken into consideration. Mechanism is driven electro-hydraulically or electrically. Different researchers have worked out on the kinematics and dynamics of the Stewart platform [9]–[12]. But did not considered much on the actuation dynamics although Stewart platform are used very often in engineering application but with small efforts of research in the field of actuation and control which is full dynamics of the platform.

2.2 Ball Screws

The ball screw research demands in application in engineering fields increased due to its high positional accuracy and accurate degree measurement. The ball screw is motion and force transfer device and the main advantages of the ball screw are its high precise movement, high speed and high efficiency. In the feed drive mechanism it is most suitable for machine tools precise movement between the tools and the work piece. It can also be used in loading and leveling of the object on the 6 DoF platforms. For high speed and high efficiency of ball screw, high speed translational ball screw is used with accurate degree measurement and with preload to increase stiffness of the ball screw mechanism.

But with preload application increases the friction between the ball screw and nut of the mechanism and heat generation in the mating part which require high torque value of servo motor. So it is suggested preload in the mechanism which requires low torque of driving, minimize slip to avoid friction and maximize the efficiency of loaded ball screw shaft.

The ball screw system is close loop mechanism and we don’t know exactly what happens in the mechanism during motion of system. Finite element analysis are used to calculate the kinetic parameters of the dynamics and kinematics of the ball screw mechanism and built performance parameter such as slide and role ratio, friction coefficient and efficiency of ball screw mechanism. The kinetics involve in the ball bearing is same to the ball screw mechanism therefore analytical calculation of the ball bearing can we used for ball screw kinetics.

Harris [13, 14] analyzed the skidding problem occurs in the glide ways and the balls produce distress of the rolling surface which is then ladder for destruction. In this model Harris focuses mainly on the contact angles of ball screw and angular velocity of the mechanism motion transformation.

Lin et al. [15] presented the mechanism which describes motion of the balls in the ball screw system and behavior of slipping between balls and other elements in contact of the balls of mechanism. Conditions of slip occur in the system is outcome and conclusion of this study and other previous study nullify the slip in the raceway of the system which is not realistic.

Lin et al. [18] introduced three ways for finding mechanical efficiency of ball screw mechanism by providing close form solution for motion of ball screw for mechanical efficiency. Optimum design to be constructed for the system where friction coefficient, contact angles and normal forces generated by a ball nut and ball screw are supposed to be same, the drag force was not measured in force balance.

Wei, Jen fin Lin [20] presents theoretical kinematics analyses of the of a double-cycle single-nut ball screw. For adjustment in the preload can be achieved by providing an offset on the two ball tracks center pitch. The critical load is larger than the preload become the edge between the axial load regions. The angle of contact in the nut and on two ball tracks varies with change on the axial load applied to the system. A. Kamalzadeh, Kerkorkmaz [21] presents a control strategy relating precision for ball screw mechanism, achieving high positioning accuracy in ball screw mechanism. Modeling of the axial vibration can be compensated in the control law according to this law high positioning bandwidth realization is enabled. Lead errors, which are repeatable, are also modeled and compensated in the control law. Effectiveness of the proposed strategy is demonstrated in high speed tracking experiments Lead errors, arising from imperfections of the screw, are removed from the loop by offsetting their effect from the command trajectory and position feedback signals. After test conducted on a ball screw drives, where a linear positioning accuracy of 2.6 um has been maintained while traversing the axis at 1000 mm/s feed with 0.5 g acceleration.Kim, Chung [24] has published data regarding the friction of the ball screw shaft mechanism in frequency domain. Nonlinear function which Kim has described includes Coulomb, and viscous friction and static friction. Velocity control loop of limit cycle analysis has been used in finding out friction elements .this method has been verified with finite element simulation and experimental variation of the ball screw mechanism.

Xian-chun, Song Jian [26] presents Contact stress analysis on the high-speed double-nut ball screw, contact deformation has been derived and got the formulas & some main factors of which could influence axial stiffness of ball screws, such as preload, helix angle, contact angle, etc. after this test for influence on ball screws caused by these factors and the corresponding related curves, which can provide theoretical support for the reasonable design on new high-speed ball screws. Contact angle and preload have great impact on axial contact stiffness of the double-nut ball screw, the stiffness of ball screws can be improved obviously by increasing the contact angle or the proper enlargement of preload.

Ball screw mechanism with different contact angles and elastic deformation of kinematic analysis was presented by Wei and Lin [27]. Wei et al. [31] presented Analytical modeling of ball screw with given preload and applied with lubrication, and finite element simulation leads to the mechanical efficiency of the experimental results. When axial load reaches preload then the mechanical efficiency decreases at a faster rate. Axial load value should be higher than the preload of ball screw by three times approximately. [30] To keep mechanical efficiency value high enough as required.

Geometry and friction in the contacts area varies with the speed that screw is rotating with high speed or low speed, generally the dividing point of the two sub regions between high and low speed is 1000 rpm .centrifugal force of balls increased as in the case where ball screw is operating at very high speed [27] and the starting slip of the balls between raceway starts.

The centrifugal force of balls in mechanism where ball screw is moving at very high speed can be determined from the speed of ball screw rotation and the axial load of ball screw and nut [27]. Normal force of ball screw contact varies inversely with the centrifugal force, with increase in the centrifugal force the normal force of ball and screw at contact point decreases. The contact angle will change many degrees and other parameters affected too with increase in centrifugal force i.e. Force of friction, normal force at contact and slipping of balls in the contact point of ball screws.

In this study of kinematics of ball screw preloaded having single nut double cycle was established with condition of lubrication between parts. Preloading can be done by offsetting the pitch of nut from centre between two ball tracks. Axial load to be applied in the end of ball screw with left side of nut which is parallel to screw axis. In this study contact angle were also verified with the between left and right nut and difference is measured and the difference of contact angle with respect to axial load as output was also presented.

The mechanical efficiency measured by the presented study of ball screw has been compared with the experimental results. The analytical calculation value for mechanical efficiency of mechanism with oil lubrication is in great agreeable condition with the experimental data [31.].

The outcome of the study is that when axial load reaches the preload value then the mechanical efficiency is decreasing and if the axial load is constant and increasing rotational torque, speed of the ball screw actuator then in this case to the mechanical efficiency decreases.

After conducting experimental testing of ball screw operating at high axial load condition and very low speed of rotation of the ball screw shaft, as a result Nihei et al measured mechanical efficiency of ball screw and friction torque.

After measurement it was observed that the mechanical efficiency decreasing when axial load approach the preload value and little increase was observed in the asymptotic load value when ball screw is operating in the high load region.

Difference is less than 3% after comparison value of mechanical efficiency we get from experiments and analytical calculation at a condition of high axial load regions and that is why this modal can be used to measure the ball screw performance.

When the ball screw is moving at very high rotational speed and mechanism normal force is relatively smaller then the centrifugal force in this case is of vital importance. This behavior has been studied by Wei and Lin [19].

The outcome of the centrifugal force on equilibrium increases which causes huge variation in between the ball nut and ball-screw contact angles , these variation in values also increases with rotational velocity when the axial load reaches the preload of ball screw mechanism.

The contact angles between the ballnut and ball screw is directly proportional of axial load.

2.3 Expert System

Engineering design involves the application of technical knowledge in a structured way to produce the definition of a product that meets specific needs. As the industrial requirements changes with time therefore there is enormous pressure on the designer to change the systems as per demands e.g. technology change, large intricate mechanism and working with different punitive workmanship is presented in [1]

Two approaches have been proposed and investigated to help designers make good choices through- out the process of design, namely virtual reality simulation and rapid prototyping [1]. This approach is called as Virtual Prototyping (VP). This approach is numerical type connected with fast and low-cost computer power processing, helps change and optimization of the design variable for the designed components in a very rapid, profitable and efficient way [2].

The other approach is called Collaborative Product Development (CPD) is also widely studied by the research officials to advance the decision-making of design engineers. A distributed CPD system is chiefly useful for modern product improvement which is being done more often by geographically and temporally dispersed design squads [3]

Existent CPD systems mainly focus on supportive activities as common access of data and collective meditation and design of components and assemblies [4]. Though, few of these tools helps the collective work in modeling and simulation to estimate the performance of proposed idea and solutions. The research questions raised in this research are; CMS can be implemented for what kind of simulations.[5]. These arise two further questions: how the integration of tools to be supported and how models to be re-used the and previous designs create codes.

T.S. Chan, H.K. Chan [40] presents homework of using model in development a new Printed Circuit Board (PCB) industrial system. SIMPROCESS software was used to implement Simulation models, with the main idea of visual interactive simulation. On the other hand, models procedure is assisted by the expert system package VP-EXPERT.

The function and configuration of information of aided design system for ship power subsystem automation is presented in [41] by Ryszard Arendt. Data telling collaborating mode to an expert system, and then creation rules evaluate correctness to the substance and form of the design. Arendt [41] proposed knowledge representation of designed ship power subsystem, and a choice of component model structures of sub systems, which enable an application of production rules for simulation models creation and design correctness evaluation are shown.

Masmoudi, Hédi Chtourou [42] presents the development of an enhanced version of the Model Expert System based Method (SESA) previously used for solving industrial system (MS) machine sizing problem. [42] Presents an improved version of the SESA connecting with Expert system .Kowalski, Ryszard Arendt [43] presents functions and arrangements of an expert system for aided design of ship systems mechanization. The method was build up on basis of design process analysis of ship system mechanization. Vecchio, Towill [44] proposed the design production and delivery systems in terms of high level, long time business logistic strategy as opposed to competent day to day individual process operation.

Chapter 3 : Research Methodologies

In research methodology we will need to do experiments, modeling of 6 DOF platform, ball screw and mechanical structure of the system and Finite element analysis in ANSYS workbench. After this we will develop a database in Excel visual basic which will evaluate data changes obtained from the respective analysis and experiments of the system.

3.1 Research Methodology

Flow chart for the research methodology has been given below with detailed picture for research & development of expert system for ball screw actuator mechanism.

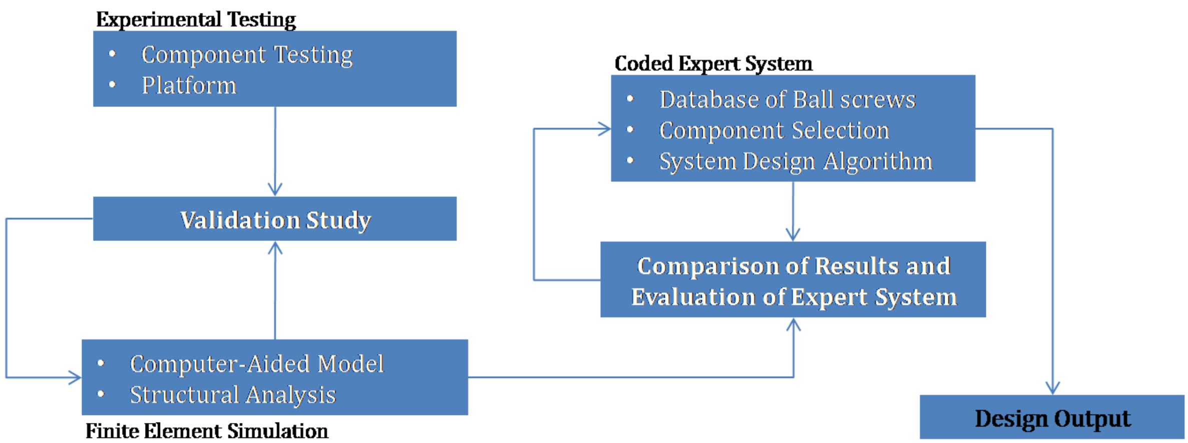


Figure 3.1: Research methodology Flow Chart

3.2 Experimental Testing

We have to test the ball for all the possible reason due to which the ball screw may fail, most probably there are five reason with which the ball screw may fail.

3.2.1 Dynamic Loading Excessive Torque

Great number of rotating ball screw fail under fatigue failure of material while rotating at high speed several times. Selecting a ball screw with material of having more capacity to dynamic load, failure can be avoided. Or we can eliminate failure of dynamic loading by minimizing the number of revolutions and reducing the load.

Ball screws should not be loaded all the time to their static strength. A practical strength for a ball screw, which may be stubborn for large travel, is almost 10% of its dynamic strength.

3.2.2 Static Load

Increasing the static load capacity on the ball screw undergoes sudden and plastic destruction to the mechanism actuators due to brinelling of balls and ball tracks and prevents the ball screw from any further normal operation.

3.2.3 Ball screw buckling

Buckling is failure of the ball screw under compressive force, the value Buckling depends on mounting pattern and free length of the ball screw. In design machinery, simple formula used for calculation is the *Euler* formula or equation. Other than this we can use nonlinear finite element simulation to predict accurately the failure of the ball screw and involved a lot of mathematical calculations.
These methods are extensively used in aerospace industry, where due to weight limitations excess safety margins are not practical.

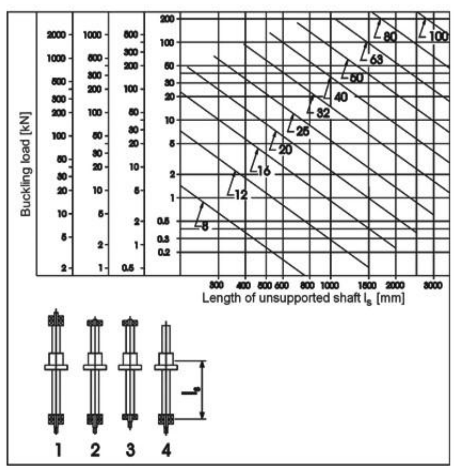


Figure 3.2: Buckling Analysis Chart

Due to log scale on left side of the graph, data for screws with longer length and with large diameter is very difficult to read. The equation below is preferred to use in this situation:

$$P_B = \frac{m \cdot d_N^4}{l_s^2} \cdot 10^4 [N]$$

$$F_{max} = 0{,}5 \cdot P_B$$

3.2.4 Failure of Nut or Shaft Body

Before the static load occur in the mechanism the nut of ball screw and ball screw itself fails and disturb the mechanism load data.

3.2.5 Radial Loads

Radial load always occur due to misalignment of slide ways or guide ways which should not be neglected and should be minimized. Under normal conditions, a radial load is lower as 6 % of the min axial load then problems will be less for damage to occur.

3.3 Coded Expert System

Coded expert system is knowledge base Expert System (ES). Knowledge base system is computer program that uses the expert's knowledge to help work men in executing various functions in various fields. Expert system is used for quick response and based on analytical calculations and date after experiments and FE analysis to be adjusted which improves quality and evaluates the data of the expert system after experiments.

3.3.1 Database

Developing an Expert system database for Ball screws selection criteria for the system required in Microsoft Excel visual basic for further analysis and for easy access of ball screw different parameter required for data analysis.

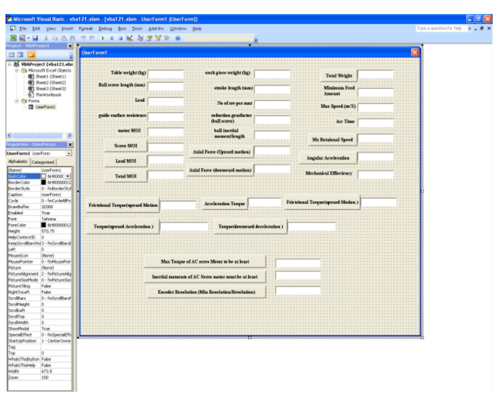


Figure 3.3: Ball screw Database Form in Excel Visual basic

3.3.2 Selection of Ball Screw Parameters

Procedure for selecting parameters for ball screw is given below which gives high efficiency and less expensive etc.

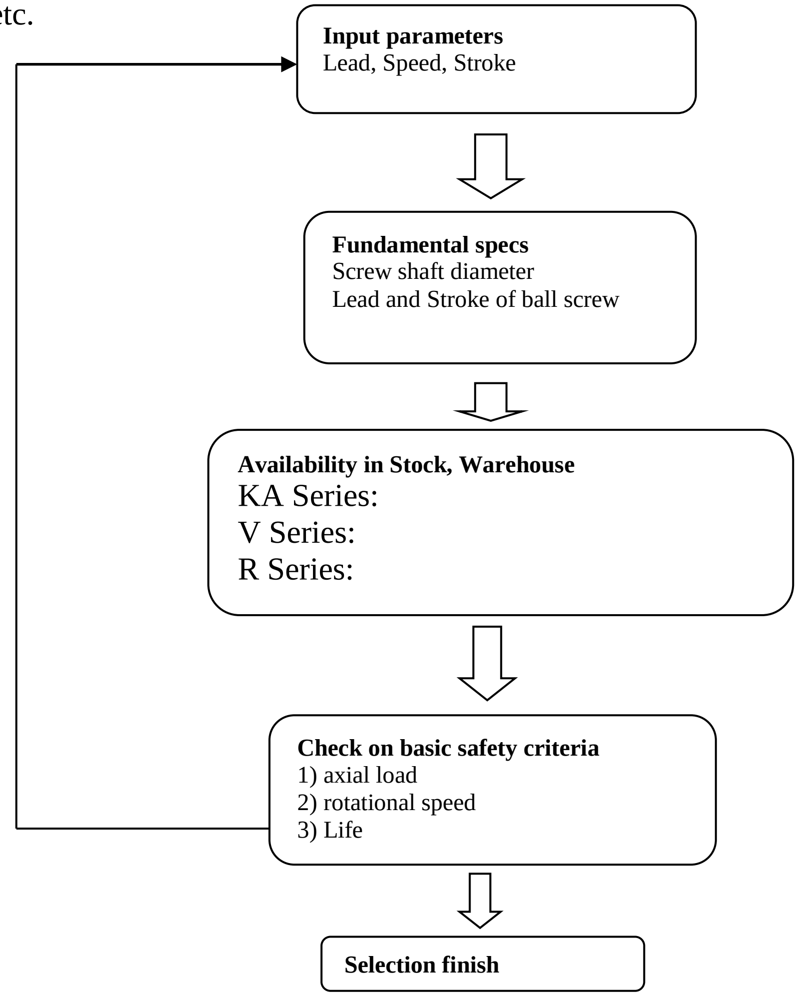


Figure 3.4: Flow Chart for Selection of Ball Screw

3.4 Finite Element Simulations

3.4.1 Modeling of System

Computer-Aided Modeling of the system comprising of the ball screw mechanism,6 DoF Platform & mechanical joints in the mechanical modeling software pro-engineer.

3.4.2 Structural Analysis

Modal structure analysis will be done in the Ansys workbench which will be compared with experiments data for correction and adjustments.

3.5 Validation of Results

3.5.1 Validation of Numerical Results

Validation and Comparison of FE simulation results with experimental data. If required, adjustments/corrections in CAD model/simulations models. To validate the results of FE simulations (stress analysis, modal analysis etc.), we shall resort to experimental structural testing. Results from experiments will serve as a benchmark for authentication of numerical simulations.

3.5.2 Evaluation of Coded Expert System

Once the FE-based simulation results are verified, they will then be used as a standard for evaluation of the coded expert system, and for any adjustments/corrections that may be required in the expert system. Design calculations to be performed in coded expert system and results will be Comparison of design results with verified numerical results If required, adjustments/corrections in coded expert system Optimization of design calculations and finalization of design variables.

Chapter 4 : Coded Expert System

4.1 Design of Ball Screw

Following are design specification for the ball screw shaft, balls and screw nut.

Data Available

- stroke length=508 mm
- max speed=381 mm
- open length of ball screw=758 mm
- close length=450 mm
- Axial Force=20 KN

Data required for design of Screw shaft

- Total length of shaft=758 mm
- length of threaded shaft=708 mm
- Dia of Nut=50 mm
- Length of Nut=50 mm
- Nominal Dia of shaft=25 mm.
- Ball diameter=5 mm.
- Root Dia of Shaft=19.5 mm.
- Mean Dia of shaft=25.5 mm.
- Major Dia of shaft=30.5.
- Lead=10 mm.
- Lead angle= 7.120^0.
- Distance b/w mountings=508+250= 758 mm.
- Table mass=700 kg.
- Work piece mass=1000 kg.
- Acceleration Time=0.2 sec.
- Deceleration Time=0.2 sec.
- Buckling load of Screw shaft=137.5 KN.
- Compression & Tensile load of shaft=34 KN.
- Inertial moment /unit length of screw shaft= 4×10^{-6} kg cm^2 /mm

- Backlash=0.1 mm.
- Rated Rotational speed of AC servo motor= 3000 min^{-1}.
- Reduction gear factor=1 (direct coupling).
- Frictional coefficient of guide surface μ=0.003 (Rolling).

4.2 Selection Conditions

Weight of Table m_1 =700kg

Work piece weight m_2 =1000kg

Length of stroke L_s= 508 mm

Max speed V_{max}=0.381m/s=381 mm

Max Axial Load $F_a = 19.918\ KN \sim 20\ KN$

Acceleration time t1 = 0.2s

Deceleration time t3 = 0.2s

Backlash 0.1mm

Rated rotational speed: 3,000 min-1

Ratio for Reduction gear None (direct coupling)

Guide surface Frictional coefficient μ =0.003 (rolling)

Resistance of Guide surface f=20 N (without load)

4.3 Parameter Selection

Diameter Screw shaft

Lead of the screw shaft

Lead Accuracy

Axial clearance

Screw shaft mounting method

Servo motor drive

4.3.1 Selection material of ball screw

a) Selecting Lead using expert system.

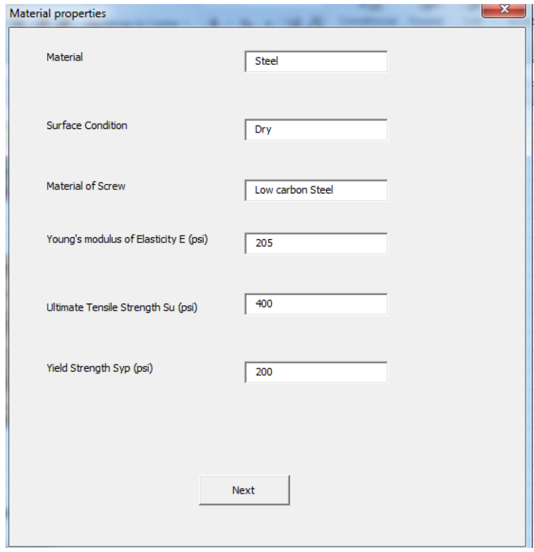


Figure 4.1: Material properties of Ball Screw shaft

b) Code for material properties of ball screw

```
Private_sub CommandButton1
UserForm3.Show
End_sub
```

4.3.2 Selecting a Screw Shaft

4.3.2.1 Assuming the Screw Shaft Length

Suppose total length of ball nut is 50 mm screw shaft end length is 250 mm then the total length of the shaft can be calculated from formula given below with the length of stroke equal to 508 mm. margin length should be 100 mm for safe movement under high load conditions.

Total length of stroke is calculated as

$$\mathrm{L_T = L_{Threaded} + L_{JEB}} \tag{4.1}$$

$$\mathrm{L_{Threaded} = L_{stroke} + L_{Nut} + L_{margin}} \tag{4.2}$$

$$L_T = L_{stroke} + L_{Nut} + L_{margin} + L_{JEB}$$

$$L_T = 508 + 100 + 100 + 50$$

The screen shaft length comes out to be 758 mm.

a) Screw shaft length selection using expert system.

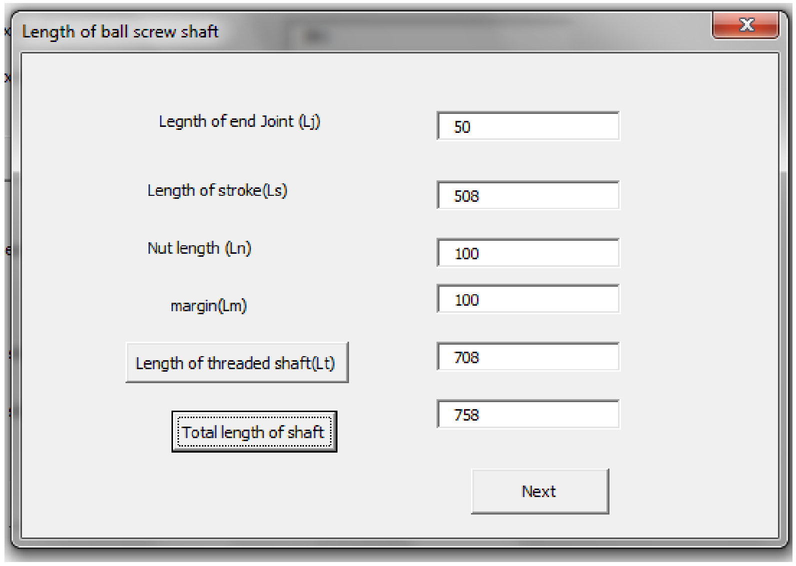


Figure 4.2 : Length of Ball Screw Shaft.

b) Code for screw shaft length selection.

```
Private_sub CommandButton1
Dim a, b, c,
a = Textbox2.text
d = Val(a) + Val(b) + Val(c)
Textbox5.text = d
End_sub
```

```
Private_sub CommandButton2
Dim a, b, c, d As Double
b = Textbox5.text
c = Val(a) + Val(b)
Textbox6.text = c
End_sub
```

```
Private_sub CommandButton3
UserForm5.Show
End_sub
```

4.3.2.2 Lead Selection for ball screw

The servo motor rated rotational speed is given as 3,000 min^{-1} and the maximum speed of rotation of ball screw is 0.381 m/s, the Ball Screw lead can be calculated as follows:

$$\text{Lead} = \frac{V_{max}}{N_{max}} \tag{4.3}$$

$$Lead = \frac{381 \times 60}{3000} = 7.62 \sim 8\,mm$$

Therefore, the lead for the ball screw will be of 10mm or longer.

4.3.3 Screw Shaft Diameter Selection

Screw shaft diameter of the loaded ball screw shaft is selected from lead of 10 mm as per given Table-4.1 are as follows.

Table 4.1: Ball Screw Combination of shaft diameter and Lead

Screw shaft outer diameter	Lead																			
	1	2	4	5	6	8	10	12	16	20	24	25	30	32	36	40	50	60	80	100
6	●																			
8		●																		
10		●			○															
12		●				○														
14			●	●																
15							●			●			●							
16				●					●											
18						●														
20				●			●			●						●				
25				●			●					●					●			
28					●															
30																		●		
32							●							●						
36							●			●	●				●					
40							●									●			●	
45								●												
50									●								●			●

●: Standard stock
○: Semi-standard stock

Shaft diameter	Lead
15mm	10mm
20mm	10mm
25mm	10mm
32mm	10mm

a) Selecting Lead using expert system.

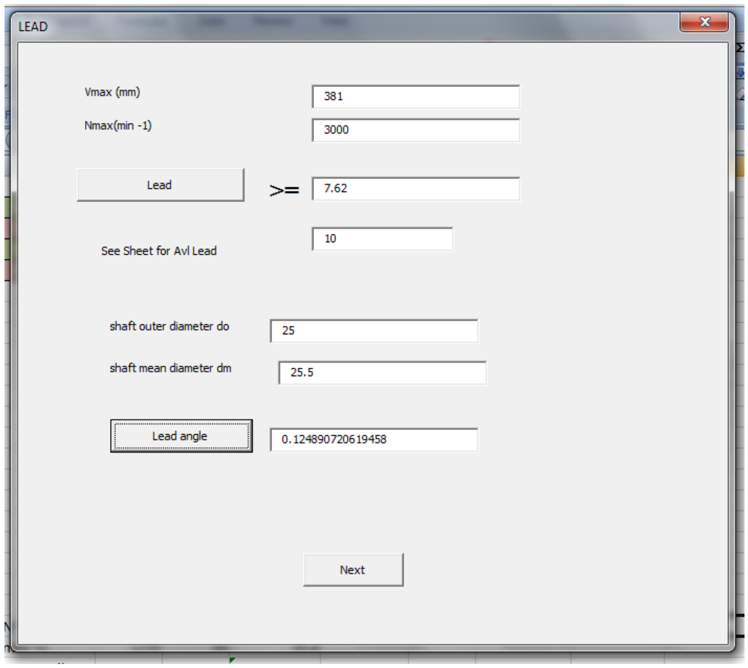


Figure 4.3: Lead of Ball Screw Shaft

b) Code for lead selection.

```
Private_sub CommandButton1
Dim a, b, c, d, e As Double
c = (Val(a) * 60) / Val(b)
Textbox3.text = c
End_sub
```

```
Private_sub
a = Textbox4.text
b = Textbox6.text
c = Val(a) / (3.14 * Val(b))
Textbox7.text = c
End_sub
```

```
Public Function Ctan(ByVal angle As Double)
Math.tan (Textbox7.text)
End Function
```

Determining root diameter of the screw shaft the distance between the mountings (l_s)of ball screw in the mechanism which is given as under:

$$\mathrm{l_a} = \mathrm{L_s} + \mathrm{L_n} + \frac{\mathrm{L_{JEB}}}{2} \tag{4.4}$$

$$l_a = 508 + 100 + 25 = 633\ mm$$

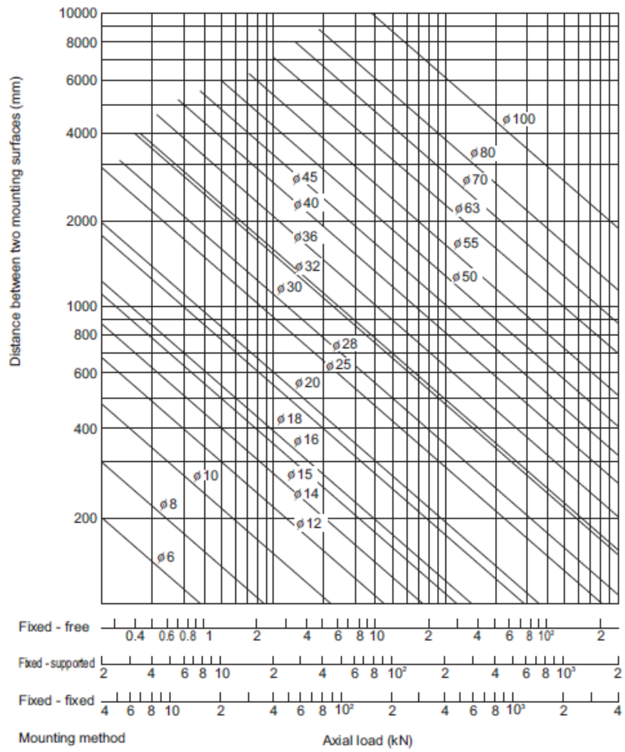


Figure 4.4 : Permissible compressive tensile load diagram

By selection fixed –supported type of mountings configuration we can conclude value of root diameter of the shaft from max axial load of 20 KN and distance b/w mountings as 633 mm.

The root diameter of the shaft comes out to be 20.5 approximately. It means that we should use value more than 21 for shaft diameter. Shaft diameter of 25 mm and a 10 mm lead of ball screw is selected for ball screw design.

a) Root Diameter Selection using expert system.

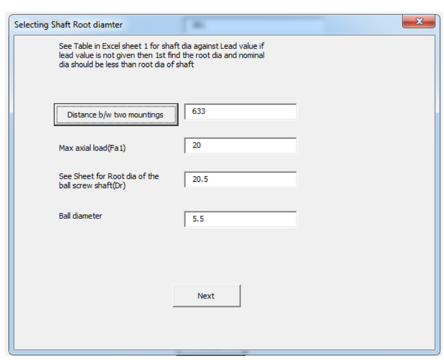


Figure 4.5: Shaft Diameter root diameter of Ball Screw

b) Code for Root diameter selection.

```
Private_sub CommandButton1
Dim a, b, c, d As Double
a = UserForm4.textbox2.text
b = UserForm4.textbox1.text
d = UserForm4.textbox3.text
c = Val(a) + (Val(b) / 2) + Val(d)
Textbox1.text = c
End_su
```

```
Private_sub CommandButton3
userform2.Show
End_sub
```

4.4 Studying the required Axial Load

4.4.1 Maximum Axial Load Calculation

Resistance Guide surface	f=20 N (without load)
Table weight	m_1 =700 kg
Work piece weight	m_2=1000 kg
Max speed	V_{max}=0.381 m/s
Time for Acceleration	t_1 = 0.2s

Values required are calculated as follows.

Acceleration for load application

$$\alpha = \frac{V_{max}}{t_1} \tag{4.5}$$

$$\alpha = \frac{.381}{0.2} = 1.905\ m/s$$

Acceleration During upward motion:

$$F_{a1} = (m_1 + m_2) \times g + f + (m_1 + m_2) \times \alpha \tag{4.6}$$

$$F_{a1} = (700 + 1000) \times 9.8 + 20 + (700 + 1000) \times 1.905 = 19918.5N$$

Acceleration during uniform motion (upward):

$$F_{a2} = (m_1 + m_2) \times g + f \tag{4.7}$$

$$F_{a2} = (700 + 1000) \times 9.8 + 20 = 16680\ N$$

Deceleration During upward motion:

$$\mathrm{F_{a3}} = (\mathrm{m_1} + \mathrm{m_2}) \times \mathrm{g} + \mathrm{f} - (\mathrm{m_1} + \mathrm{m_2}) \times \alpha \tag{4.8}$$

$$F_{a3} = (700 + 1000) \times 9.8 + 20 - (700 + 1000) \times 1.905 = 13441.5\ N$$

Acceleration During downward motion:

$$\mathrm{F_{a4}} = (\mathrm{m_1} + \mathrm{m_2}) \times \mathrm{g} - \mathrm{f} - (\mathrm{m_1} + \mathrm{m_2}) \times \alpha \tag{4.9}$$

$$\mathrm{F_{a4}} = (700 + 1000) \times 9.8 - 20 - (700 + 1000) \times 1.905 = 13401.5\mathrm{N}$$

Acceleration During uniform motion (Downward):

$$\mathrm{F_{a5}} = (\mathrm{m_1} + \mathrm{m_2}) \times \mathrm{g} - \mathrm{f} \tag{4.10}$$

$$\mathrm{F_{a5}} = (700 + 1000) \times 9.8 - 20 = 16640\ \mathrm{N}$$

Deceleration During downward motion:

$$\mathrm{F_{a6}} = (\mathrm{m_1} + \mathrm{m_2}) \times \mathrm{g} - \mathrm{f} + (\mathrm{m_1} + \mathrm{m_2}) \times \alpha \tag{4.11}$$

$$F_{a6} = (700 + 1000) \times 9.8 - 20 + (700 + 1000) \times 1.905 = 19878.5\ N$$

Thus, maximum value of axial load calculated on the Ball Screw is given as below:

$F_{a\,max} = F_{a1}$ = 19918 N

a) Selecting Max Axial Load using expert system.

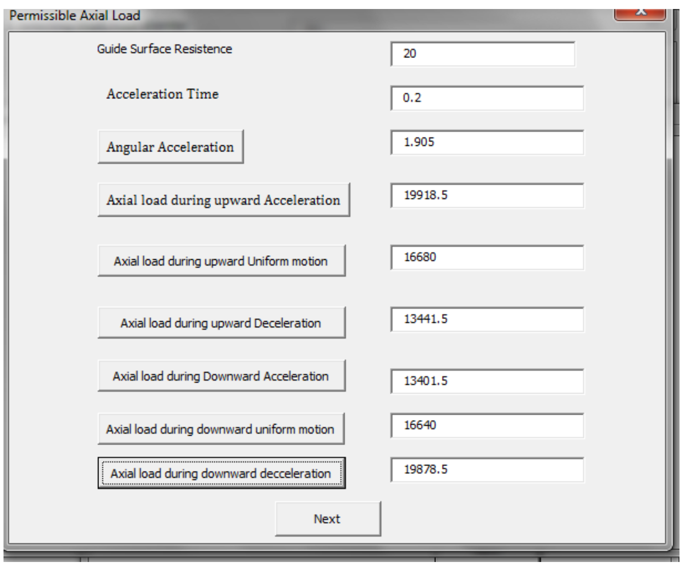


Figure 4.6: Permissible Axial Load of Ball Screw

b) Code for Max Axial Load selection.

```
Private_sub CommandButton1
Dim a, b, c, d As Double
a = UserForm3.textbox1.text
b = Textbox1.text
Textbox2.
End_sub
```

```
Private_sub CommandButton2
Dim a, b, c, d
a = userform2.textbox3.text
b = Textbox9.text
c = Textbox2.text
d = (Val(a) * 9.8) + Val(b) + (Val(a) * Val(c))
Textbox3.text = d
End_sub
```

```
Private_sub CommandButton3
Dim a, b, c, d
a = userform2.textbox3.text
b = Textbox9.text
d = (Val(a) * 9.8) + Val(b)
Textbox4.text = d
End_sub
```

```
Private_sub CommandButton4
Dim a, b, c, d
a = userform2.textbox3.text
b = Textbox9.text
c = Textbox2.text
d = (Val(a) * 9.8) + Val(b) - (Val(a) * Val(c))
Textbox5.text = d
End_sub
```

```
Private_sub CommandButton5
Dim a, b, c, d
a = userform2.textbox3.text
b = Textbox9.text
c = Textbox2.text
d = (Val(a) * 9.8) - Val(b) - (Val(a) * Val(c))
Textbox6.text = d
End_sub
```

```
Private_sub CommandButton6
Dim a, b, c,
a = userform2.textbox3.text
b = Textbox9.text
d = (Val(a) * 9.8) - Val(b)
Textbox7.text = d
End_sub
```

```
Private_sub CommandButton7
dim a, b, c, d
```

```
a = userform2.textbox3.text
b = Textbox9.text
c = Textbox2.text
d = (Val(a) * 9.8) - Val(b) + (Val(a) * Val(c))
Textbox8.text = d
End_sub
```

```
Private_sub CommandButton8
UserForm7.Show
End_sub
```

4.5 Ball Screw Shaft Buckling Load

Factor for Buckling as per mounting method η2=10

Since the boundary condition for the ball screw shaft between the nut and the loaded platform, in which buckling is to be determined is "fixed-pinned supported:"

Modulus of Elasticity =E = 205 GPA

Length of shaft = L=758 mm

Screw shaft Nominal diameter of the shaft =25 mm

Moment of Inertia of Shaft

$$I = \frac{\pi \times d^4}{64} = 19165.039 \tag{4.12}$$

Screw-shaft thread minor diameter or the root diameter is $d_r = 20.5\ mm$

Buckling factor =K= 0.7

$$\mathrm{P_{buckling}} = \frac{\pi^2 \mathrm{EI}}{\mathrm{L^2K^2}} = 137.59\ \mathrm{KN} \tag{4.13}$$

$$P_{buckling} = 137582\ N \sim 137.58\ KN$$

4.6 Compressive and Tensile Load

Compressive and tensile of ball screw shaft is given by formula given as

$$\mathrm{P}_{\mathrm{Comp}\ \&\ Tesile} = \sigma \times \frac{\pi}{4} \times \mathrm{d_r^2} = 48.75\ \mathrm{KN} \tag{4.14}$$

$$P_{Comp\ \&\ Tesile} = 48749\ N \sim 48.75\ KN$$

Ball screw Buckling, compressive and tensile load equal or greater than axial load max value. As the values for buckling and compressive and tensile stresses are more than the axial load therefore, ball Screw with given lead, screw shaft diameter can be used without any problem.

a) Selecting Buckling load, Compressive & Tensile load using expert system.

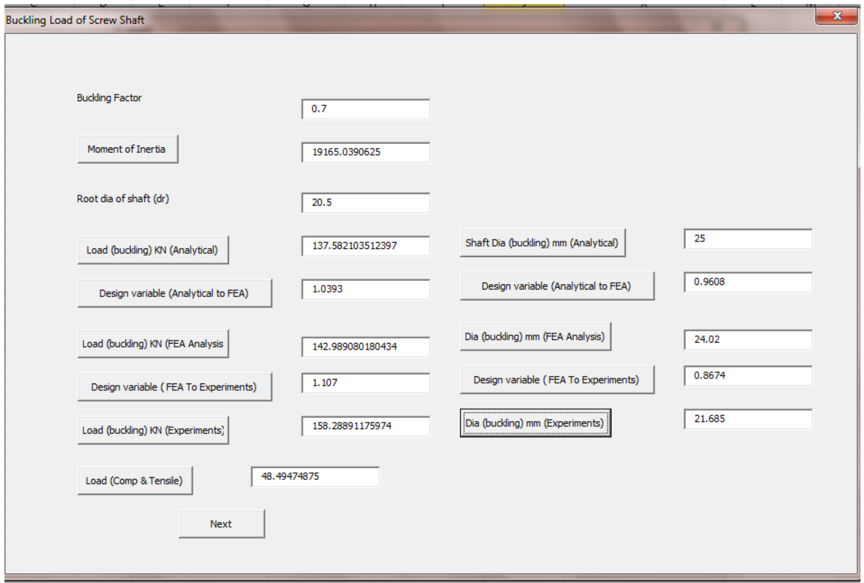


Figure 4.7: Buckling, compression and Tensile Load of Ball Screw

b) Code for Buckling, Compressive& Tensile selection.

```
Private_sub CommandButton1
Dim a, b, c, d, e As Double
b = Textbox1.text
e = UserForm4.textbox6.text
d = (Val(a) * Val(c) * 9.859) / ((Val(b) ^ 2) * (Val(e) ^ 2))
Textbox4.text = d
End_sub

Private_sub CommandButton2
Dim a, b, c, d As Double
a = Textbox3.text
d = 116 * (Val(a) ^ 2) / 1000
Textbox5.text = d
End_sub

Private_sub CommandButton3
UserForm14.Show
End_sub

Private_sub CommandButton4
Dim a, b, c, d As Double
a = UserForm3.textbox5.text
c = (3.14 * Val(a) ^ 4) / 64
Textbox2
End_sub

Private_sub Textbox2_Change()
Dim a, b, c, d As Double
a = UserForm3.textbox5.text
```

```
b = (3.14 * Val(a) ^ 4) / 64
Textbox2.text = b
End_sub
```

```
Private_sub Textbox3
Textbox3.text = UserForm5.textbox3.text
End_sub
```

4.7 Studying the factor of safety of fatigue loading.

Length of shaft = 758 mm

Screw-shaft thread minor diameter or the root diameter is $d_r = 20.5\ mm$

Screw shaft Nominal diameter of the shaft =25 mm

Ultimate tensile strength of the shaft = 400 MPa

Fluctuating tensile load of shaft varies between: 0 to 20KN

Area of the screw shaft =

$$\frac{\pi}{4} \times d^2 = \frac{\pi}{4} \times 25^2 = 490.625\ mm^2 \tag{4.15}$$

Surface finish factor =k_a= 0.85

Size factor = k_b=1

Load factor=k_c= 0.85

Temperature factor= k_d= 1

Fatigue stress concentration factor k_f= 1.85

Maximum stress

$$\sigma_{max} = \frac{Load_{max}}{Area} = \frac{20\ KN}{490.625} = 40.76\ \frac{KN}{mm^2} \tag{4.16}$$

Minimum stress

$$\sigma_{min} = \frac{Load_{min}}{Area} = \frac{0\ KN}{490.625} = 0\ \frac{KN}{mm^2} \tag{4.17}$$

Alternating stress

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{40.76 - 0}{2} = 20.38\ \frac{KN}{mm^2} \tag{4.18}$$

Mean stress

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{40.76 + 0}{2} = 20.38\ \frac{KN}{mm^2} \tag{4.19}$$

Endurance limit

$$S'_e = 0.5 \times S_{ut} \tag{4.20}$$

$$S'_e = 0.5 \times 400 = 200 \frac{KN}{mm^2}$$

Endurance limit

$$S_e = \mathrm{K_a K_b K_c K_d S'_e} \tag{4.21}$$

$$S_e = 0.85 \times 1 \times 0.85 \times 1 \times 200 = 144.5\ \mathrm{KN/mm^2}$$

Required factor of Safety for the mechanism using Goodman interaction line is given as:

$$\frac{1}{N_f} = \frac{K_f \sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} \tag{4.22}$$

$$FOS = \frac{\sigma_a @ N_{desired}}{\sigma_{a1(design)}} \tag{4.23}$$

$$\frac{1}{N_f} = \frac{1.85 \times 40.76}{144.5} + \frac{40.76}{400}$$

$$N_f = 3.2$$

a) Selecting Fatigue factor of safety using expert system.

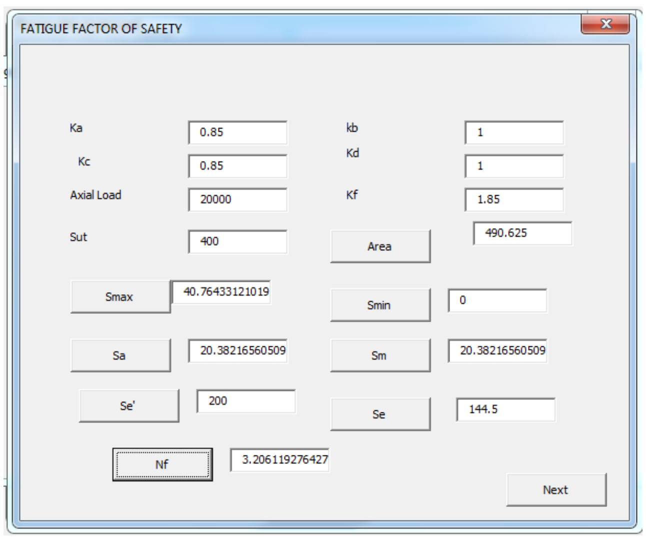


Figure 4.8: Fatigue Factor Safety of Ball Screw.

b) Code for Fatigue factor of safety calculation.

```
Private_sub CommandButton1
Dim a, b, c As Double
a = UserForm3.textbox5.text
c = (3.14 / 4) * Val(a) * Val(a)
Textbox8.
End_sub
```

```
Private_sub CommandButton2
Dim a, b, c As Double
```

```
a = Textbox5.text
b = Textbox8.text
c = Val(a) / Val(b)
Textbox9.
End_sub
```

```
Private_sub CommandButton5
Dim a, b, c As Double
a = Textbox9.text
b = Textbox10.text
Textbox12.text = c
End_sub
```

```
Private_sub CommandButton7
Dim a, b, c, d, e, f As Double
a = Textbox1.text
f = Textbox13.text
c = Val(a) * Val(b) * Val(d) * Val(e) * Val(f)
Textbox14.text = c
End_sub
```

```
Private_sub CommandButton8
Dim a, b, c, d, e, f, g, h As Double
a = Textbox6.text
e = Textbox12.text
f = Textbox14.text
Textbox15.text = c
End_sub
```

```
Private_sub CommandButton9
UserForm8.Show
End_sub
```

4.8 Required Rotational Speed of Motor

4.8.1 Maximum Rotational Speed

Ball Screw diameter: 20mm;

Ball screw lead: 10mm

Max speed Vmax= 0.381 m/s

Maximum rotational speed of motor is given as

$$N_{max} = \frac{V_{max} \times 60}{Lead} \tag{4.24}$$

$$N_{max} = \frac{381 \times 60}{10} = 2286\ min^{-1}$$

4.8.2 Motor Torque

4.8.2.1 Motor Torque Due to an External Load

Torque during uniform motion (upward) :

$$T_1 = \frac{F_{a2} \times Lead}{2 \times \pi \times \eta} \tag{4.25}$$

$$T_1 = \frac{F_{a2} \times 10}{2 \times \pi \times .9} = 29511.67\ Nmm$$

During downward uniform motion:

$$T_2 = \frac{F_{a5} \times Lead}{2 \times \pi \times \eta} \tag{4.26}$$

$$T_2 = \frac{F_{a5} \times 10}{2 \times \pi \times 0.9} = 29440.91\ Nmm$$

a) Selecting Max Rotational speed & Frictional Torque using expert system.

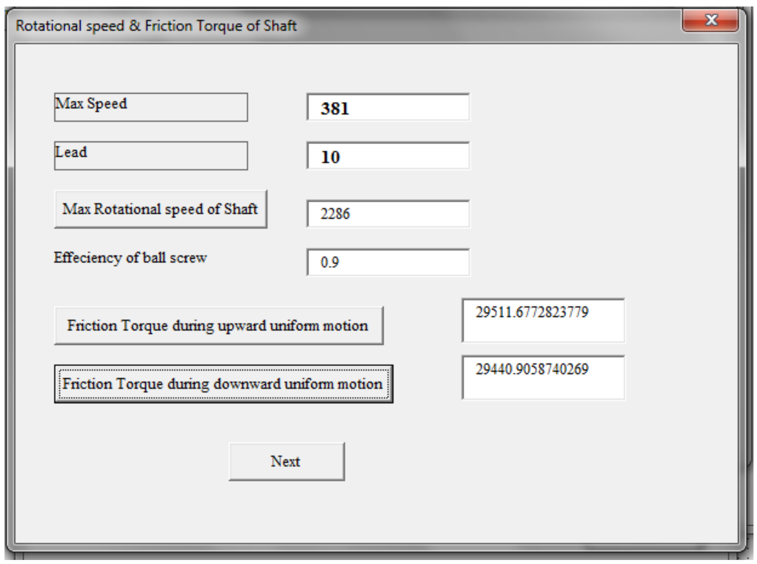


Figure 4.9: Rotational Speed and Frictional Torque of Ball Screw.

b) Code for Max Rotational speed & Frictional Torque selection.

```
Private_sub CommandButton1
dim a, b, c, d As Double
a = Textbox1.text
d = (Val(a) * 60) / Val(b)
Textbox3.text = d
End_sub
```

```
Private_sub CommandButton2
Dim a, b, c, d As Double
a = UserForm6.textbox4.text
c = Textbox4.text
d = (Val(a) * Val(b)) / (2 * 3.14 * Val(c))
Textbox5.text = d
```

```
End_sub
```

```
Private_sub CommandButton3
dim a, b, c, d As Double
a = UserForm6.textbox7.text
c = Textbox4.text
d = (Val(a) * Val(b)) / (2 * 3.14 * Val(c))
Textbox6.text = d
End_sub
```

```
Private_sub CommandButton4
UserForm9.Show
End_sub
```

```
Private_sub Textbox1
Textbox1.text = UserForm3.textbox1.text
End_sub
```

```
Private_sub Textbox2
Textbox2.text = UserForm3.textbox4.text
End_sub
```

4.8.3 Acceleration Torque

Moment of inertia or 2nd moment of area:

moment of inertia per unit length of the screw shaft = 3.9×10^{-4} kg•cm²/mm,

the moment of inertia of the screw shaft with an total length of 758 mm is calculated below as.

$$J_s = 3.9 \times 10^{-4} \times 758 = 0.295 \times 10^{-4}\ \text{kg} \bullet \text{m}^2/\text{mm}$$

Load moment of inertia

$$J = (m_1 + m_2)\left(\frac{\text{Lead}}{2\times\pi}\right)^2 \times A^2 \times 10^{-6} = 4.3 \times 10^{-3} kg.\text{m}^2 \qquad (4.27)$$

$$J = (700 + 1000)\left(\frac{10}{2\times\pi}\right)^2 \times 1^2 - 10^{-6} + 0.295 \times 10^{-4} \times 1^2 = 4.3 \times 10^{-3}\text{kgm}^2$$

Angular acceleration:

$$\omega = \frac{2\times N_{max}\times\pi}{60\times t_1} \qquad (4.28)$$

$$\omega = \frac{2 \times 2286 \times \pi}{60 \times .2} = 1196.34\ \text{rad/sec}^2$$

As a result of above calculation, acceleration torque is calculated below as.

$$T_3 = (J + J_m) \times \omega \qquad (4.29)$$

$$T_3 = (J + J_m) \times \omega = \mathit{5252.030\ Nmm}$$

a) Selecting Torque required for Acceleration using expert system.

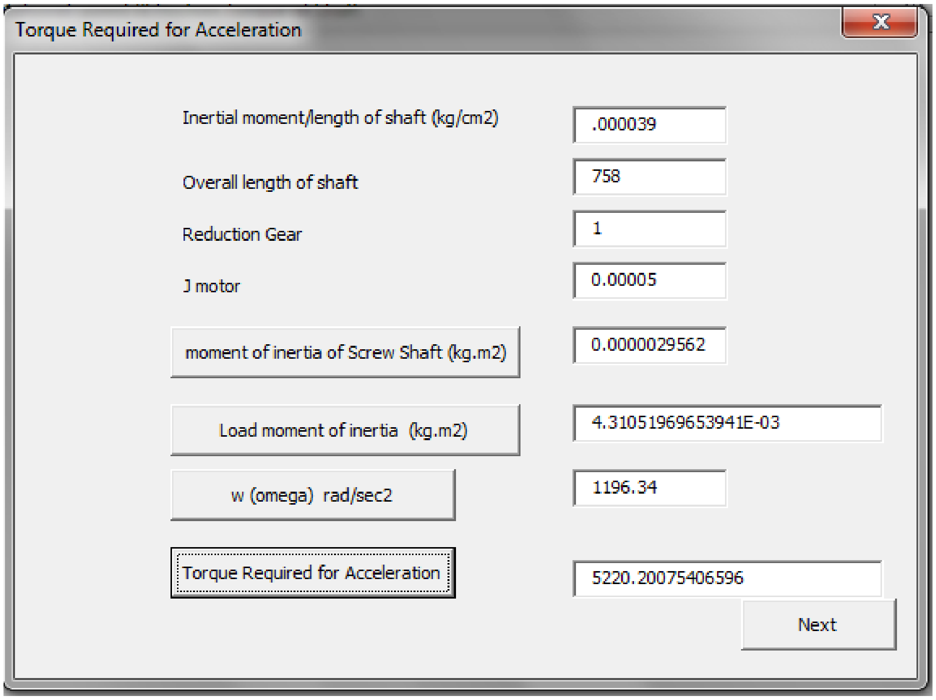


Figure 4.10: Torque required for Acceleration of Screw Shaft.

b) Code for Torque required for Acceleration selection.

```
Private_sub CommandButton1
dim a, b, c, d As Double
a = Textbox1.text
d = Val(a) * Val(b) * (Val(c) ^ 2) * 0.0001
Textbox5.text = d
End_sub
```

```
Private_sub CommandButton2
dim a, b, c, d As Double
a = userform2.textbox3.text
b = UserForm3.textbox4.text
c = Textbox3.text
d = (Val(c) ^ 2) * Val(a) * ((Val(b) / (2 * 3.14)) ^ 2) * 0.000001
Textbox6.text = d
End_sub
```

```
Private_sub CommandButton3
a = UserForm8.textbox3.text
b = UserForm6.textbox1.text
d = (2 * Val(a) * 3.14) / (Val(b) * 60)
Textbox7.text = d
End_sub
```

```
Private_sub CommandButton4
```

```
dim a, b, c, d, e As Double
a = Textbox4.text
e = Textbox7.text
d = (Val(a) + Val(b) + Val(c)) * Val(e) * 1000
Textbox8.text = d
End_sub
```

```
Private_sub CommandButton5
UserForm10.Show
End_sub
```

Acceleration torque required for ball screw shaft is given as under.

Acceleration during upward motion:

$$T_{upa} = T_1 + T_3 = 29511.67 + 5252.030 = 34763.707 \text{ Nmm} \qquad (4.30)$$

During upward uniform motion:

$$T_{upun} = T_1 = \text{T1} = 29511.67 \text{ N•mm}$$

During upward deceleration:

$$T_{upde} = T_1 - T_3 = \text{T1– T3} = 29511.67 - 5252.030 = 24259.24 \text{ Nmm} \qquad (4.31)$$

During downward acceleration:

$$T_{dwa} = T_2 - T_3 = 29440.91 - 5252.030 = 24188.87 \text{ Nmm} \qquad (4.32)$$

Deceleration during uniform motion (downward):

$$T_{dwun} = T_2 = 29440.91 \text{ Nmm}$$

Deceleration during downward motion:

$$T_{dwde} = T_2 + T_3 = 29440.91 + 5252.030 = 34692.94 \text{ Nmm} \qquad (4.33)$$

4.8.4 Motor Rotational Torque

The motor rotational torque during upward acceleration calculated above is the required maximum torque**.**

$$\mathbf{T_{max}} = \mathbf{T_{upa}} = 29511.67 \text{ N•mm}$$

Therefore, maximum rotation torque of the AC servomotor to be greater than 29511.67 N-mm.

a) Selecting Motor Torque using expert system.

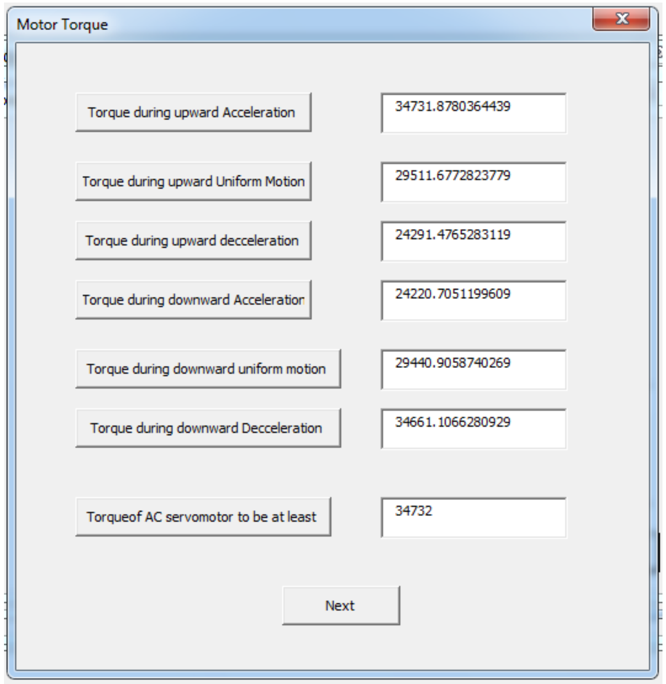


Figure 4.11: Motor Torque of Ball Screw Shaft.

b) Code for Motor Torque selection.

```
Private_sub CommandButton1
dim a, b, c, d, e As Double
a = UserForm8.textbox5.text
b = UserForm9.textbox8.text
d = Val(a) - Val(b)
Textbox3.text = d
End_sub
```

```
Private_sub CommandButton4
dim a, b, c, d, e As Double
a = UserForm8.textbox6.text
b = UserForm9.textbox8.text
d = Val(a) - Val(b)
Textbox4.text = d
End_sub
```

```
Private_sub CommandButton5
Textbox5.text = UserForm8.textbox6.text
End_sub
```

```
Private_sub CommandButton6
dim a, b, c, d, e As Double
```

```
a = UserForm8.textbox6.text
b = UserForm9.textbox8.text
d = Val(a) + Val(b)
Textbox6.text = d
End_sub
```

```
Private_sub CommandButton8
UserForm11.Show
End_sub
```

4.9 Calculating travel distance

Maximum speed V_{max}=0.381m/s=381 mm

Time for Acceleration t_1=0.2 s

Time for Deceleration t_2=0.2 s

Travel distance during Acceleration.

$$l_{1,4} = \frac{V_{max} \times t_1}{2} \times 10^3 = \frac{0.381 \times .2}{2} \times 10^3 = 38.1\text{mm} \tag{4.34}$$

Travel distance during uniform motion.

$$l_{2,5} = l_s - \frac{(V_{max} \times t_1) + (V_{max} \times t_{3)}}{2} \times 10^3 \tag{4.35}$$

$$l_{2,5} = 508 - \frac{(0.381 \times .2) + (0.381 \times .2)}{2} \times 10^3 = 431.8 \text{ mm}$$

Travel distance during deceleration.

$$l_{1,4} = \frac{V_{max} \times t_3}{2} \times 10^3 = \frac{0.381 \times .2}{2} \times 10^3 = 38.1$$

Time to be calculated by comparison 38 mm in 0.2 sec then 431.8 in 1.13333 sec

The selection requirements and the torque calculated in during upward acceleration:

T_{upa} = 34763.707 Nmm; t_1 = 0.2 s

Torque during uniform motion (upward):

$T_{upun} = T_1$= 29511.67 Nmm; t_2 = 1.133 s

During upward deceleration:

$T_{upde} = T_1 - T_3$ = 24259.24 Nmm; t_3 = 0.2 s

Torque during acceleration (downward):

$T_{dwa} = T_2 - T_3$ =24188.87 Nmm; t_1= 0.2 s

Torque during uniform motion (downward):

$T_{dwun} = T_2$ = 29440.91 Nmm; t_2 = 1.133 s

Torque during deceleration (downward):

$T_{dwde} = T_2 + T_3$= 34692.94 Nmm; t_3 = 0.2 s

4.10 Motor Moment of Inertia

The moment of inertia applied to the motor is calculated as follows

$$J = (m_1 + m_2)(\frac{Lead}{2\times\pi})^2 \times A^2 - 10^{-6} + J_s \times A^2 = 4.3 \times 10^{-3}\text{kg}\bullet m^2 \quad (4.36)$$

Generally, the motor require moment of inertia greater than one tenth of the inertial moment applied to the motor. Therefore, the moment of inertia of the AC servomotor should be equal to 4×10^{-5} kg-m2 or greater.

The selection process for ball screw selection parameter is completed.

a) Selecting Travel distance and moment of inertia using expert system.

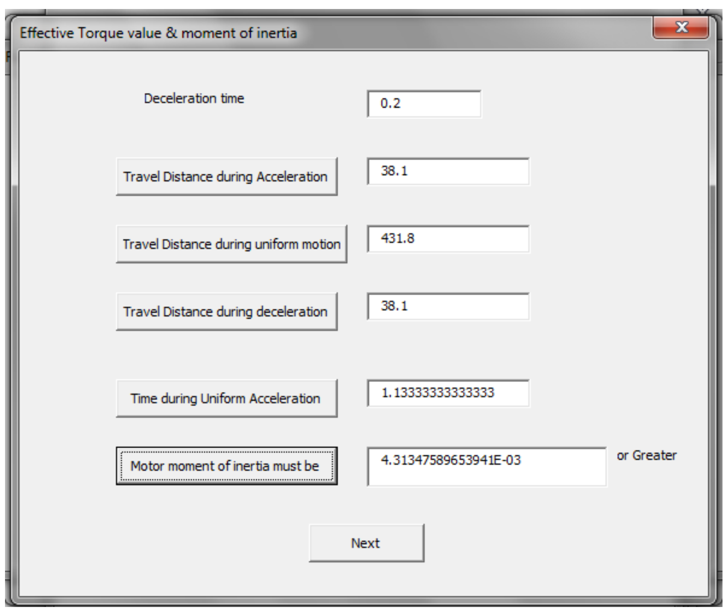


Figure 4.12 : Moment of Inertia of Ball Screw Shaft.

b) Code for Travel distance and moment of inertia selection.

```
Private_sub CommandButton2
dim a, b, c, d, e As Double
a = UserForm4.textbox2.text
b = UserForm3.textbox1.text
c = UserForm6.textbox1.text
e = Textbox6.text
d = Val(a) - (((Val(b) * Val(c)) + (Val(b) * Val(e))) / 2)
Textbox2.text = d
```

```
End_sub
```

```
Private_sub CommandButton3
dim a, b, c, d, e As Double
a = UserForm3.textbox1.text
b = Textbox6.text
d = (Val(a) * Val(b)) / 2
Textbox3.text = d
End_sub
```

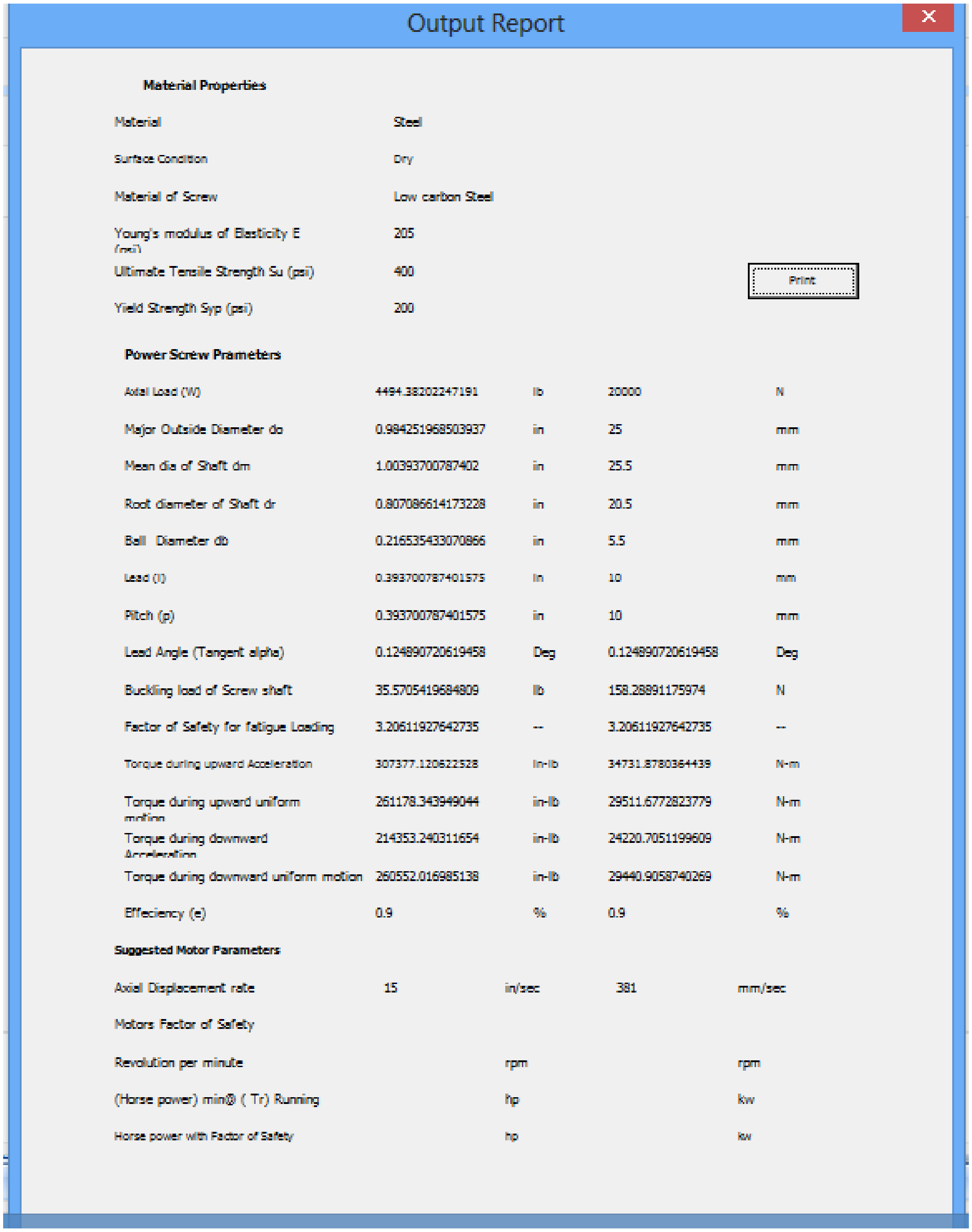
```
Private_sub CommandButton4
dim a, b, c, d, e, f, g, h, i, j As Double
a = UserForm10.textbox1.text
g = UserForm10.textbox6.text
h = UserForm6.textbox1.text
i = Textbox6.text
j = Textbox7.text
d = sqrt(((Val(a) * Val(h)) + (Val(b) * Val(i)) + (Val(c) * Val(j)) + (Val(e) * Val(h)) + (Val(f) * Val(i)) + (Val(g) *
Val(j))) / (Val(h) + Val(i) + Val(j) + Val(h) + Val(i) + Val(j)))
Textbox4.text = d
End_sub
```

```
Private_sub CommandButton5
dim a, b, c, d, e As Double
a = UserForm9.textbox5.text
b = UserForm9.textbox6.text
d = Val(a) + Val(b)
Textbox5.text = d
End_sub
```

```
Private_sub CommandButton7
UserForm13.Show
End_sub
```

4.11 Output Report of the Expert System.

a) Finding Output Report using expert system.


Output Report

Material Properties

Material	Steel
Surface Condition	Dry
Material of Screw	Low carbon Steel
Young's modulus of Elasticity E (psi)	205
Ultimate Tensile Strength Su (psi)	400
Yield Strength Syp (psi)	200

Print

Power Screw Prameters

Axial Load (W)	4494.38202247191	lb	20000	N
Major Outside Diameter do	0.984251968503937	in	25	mm
Mean dia of Shaft dm	1.00393700787402	in	25.5	mm
Root diameter of Shaft dr	0.807086614173228	in	20.5	mm
Ball Diameter db	0.216535433070866	in	5.5	mm
Lead (l)	0.393700787401575	in	10	mm
Pitch (p)	0.393700787401575	in	10	mm
Lead Angle (Tangent alpha)	0.124890720619458	Deg	0.124890720619458	Deg
Buckling load of Screw shaft	35.5705419684809	lb	158.28891175974	N
Factor of Safety for fatigue Loading	3.20611927642735	--	3.20611927642735	--
Torque during upward Acceleration	307377.120622528	in-lb	34731.8780364439	N-m
Torque during upward uniform motion	261178.343949044	in-lb	29511.6772823779	N-m
Torque during downward Acceleration	214353.240311654	in-lb	24220.7051199609	N-m
Torque during downward uniform motion	260552.016985138	in-lb	29440.9058740269	N-m
Effeciency (e)	0.9	%	0.9	%

Suggested Motor Parameters

Axial Displacement rate	15	in/sec	381	mm/sec
Motors Factor of Safety				
Revolution per minute		rpm		rpm
(Horse power) min@ (Tr) Running		hp		kw
Horse power with Factor of Safety		hp		kw

Figure 4.13 : Output Report of the selection of Ball screw in visual basic.

b) Code output report of the ball screw system.

```
Private_sub CommandButton1
UserForm13.PrintForm
End_sub
```

```
Private_sub Label104
Label104.Caption = Label106.Caption * 8.85
End_sub
```

```
Private_sub Label106
Label106.Caption = UserForm10.textbox1.text
End_sub
```

```
Private_sub Label109
Label109.Caption = Label111.Caption * 8.85
End_sub
```

```
Private_sub Label111
Label111.Caption = UserForm10.textbox2.text
End_sub
```

```
Private_sub Label114
Label114.Caption = Label116.Caption
End_sub
```

```
Private_sub Label116
Label116.Caption = UserForm8.textbox4.text
End_sub
```

```
Private_sub Label119
Label119.Caption = Label121.Caption * 8.85
End_sub
```

```
Private_sub Label121
Label121.Caption = UserForm10.textbox5.text
End_sub
```

```
Private_sub Label15
Label15.Caption = UserForm12.textbox1.text
End_sub
```

```
Private_sub Label16
Label16.Caption = UserForm12.textbox2.text
End_sub
```

```
Private_sub Label17
Label17.Caption = UserForm12.textbox3.text
End_sub
```

```
Private_sub Label18
Label18.Caption = UserForm12.textbox4.text
End_sub
```

```
Private_sub Label19
Label19.Caption = UserForm12.textbox5.text
End_sub
```

```
Private_sub Label20
Label20.Caption = UserForm12.textbox6.text
End_sub
```

```
Private_sub Label34
Label34.Caption = Label44.Caption / 1.33
End_sub
```

```
Private_sub Label35
Label35.Caption = Label45.Caption / 25.4
End_sub
```

```
Private_sub Label38
Label38.Caption = Label48.Caption / 1.33
End_sub
```

```
Private_sub Label45
Label45.Caption = UserForm3.textbox1.text
End_sub
```

```
Private_sub Label55
Label55.Caption = Label65.Caption / 25.4
End_sub
```

```
Private_sub Label56
Label56.Caption = Label66.Caption / 4.45
End_sub
```

```
Private_sub Label57
Label57.Caption = Label67.Caption / 25.4
End_sub
```

```
Private_sub Label58
Label58.Caption = Label68.Caption / 25.4
End_sub
```

```
Private_sub Label59
Label59.Caption = Label69.Caption / 25.4
End_sub
```

```
Private_sub Label85
Label85.Caption = UserForm14.textbox15.text
End_sub
```

```
Private_sub Label86
Label86.Caption = UserForm3.textbox4.text
End_sub
```

```
Private_sub Label87
Label87.Caption = Label86.Caption
End_sub
```

```
Private_sub Label88
Label88.Caption = UserForm3.textbox7.text
End_sub
```

```
Private_sub Label89
 Label89.Caption = UserForm7.textbox4.text
End_sub
```

```
Private_sub Label95
Label95.Caption = Label99.Caption * 8.85
End_sub
```

```
Private_sub Label99
Label99.Caption = UserForm10.textbox4.text
End_sub
```

Chapter 5 : Finite Element Simulations

5.1 Modeling of ball screw mechanism in Pro-Engineer

The mechanism consists of Servomotors, ball screws, nut gear assembly and mechanical joints. The mechanism consists of 06 ball screw shaft that drive six degree of freedom platform. Ball screw shaft is driven by servomotor through gear system. There is very accurate timing between the platform and the ball screw shaft that runs at controlled speed of servo motor

The mechanism flexibility has been improved as each ball screw is controlled with its own servo motor. There will be no longer a need to run the mechanism input at constant speed and theoretically, numerous ranges of stroke for different segments of input rotation could be obtained. Servo motors offer accurate control of the mechanism’s output position, velocity and acceleration controlled with feedback loops.

5.1.1 Description of Assembly

The software Pro/Engineer Wildfire 4.0 was extensively used for the design and mechanism analysis of the ball screw shaft. An accurate model of the ball screw was used for the accurately depicting the manufactured ball screw for FEA Analysis and experimental testing.

Dynamic analysis was further conducted using Pro/Mechanism in order to show that the mechanism operated under the safety factors provided of the design.

Each Actuator Assembly consist of 02 gears ,Long nut threaded at one end, Ball screw, balls in between nut and Ball screw and servo motor. The power from the servo motor is transmitted through gears to ball screw which can be used for motion of 6 DOF platforms. Full description and section view of Ball screw actuator mechanism is shown in figures below.

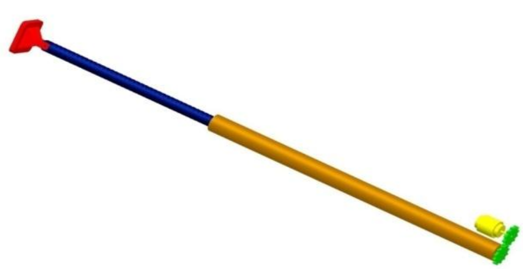


Figure 5.1: Ball screw servo motor actuator mechanism Assembly in Pro-Engineer

Figure 5.2: Ball screw Actuator Assembly section view

5.1.1.1 Modeling of Ball screw

Ball screw is modeled for accurate analysis in Pro-Engineer wildfire-4.the length of the screw shaft is 758 mm with Dia of 25 mm .mounting hole is provided on the ball screw for mounting ball screw with the mechanical joint which is then connected with the load .

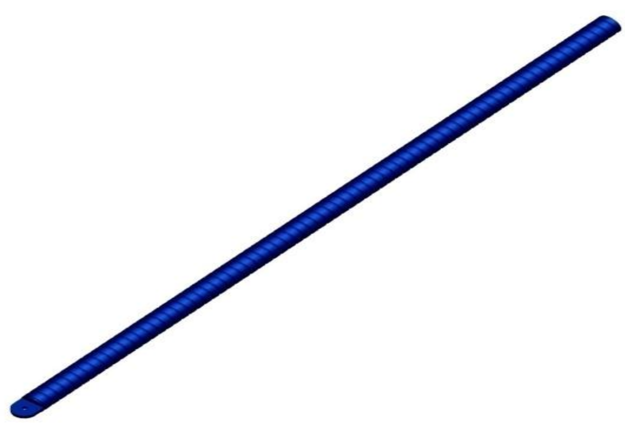


Figure 5.3: Ball screw of the actuator mechanism

5.1.1.2 Modeling of Ball nut

Ball nut is used in the mechanism is modeled in such a way that one end of the shaft is connected with the 2nd gear assembly and other end of the ball nut is connected with the ball screw. Balls are in between the screw shaft and nut of ball screw assembly which provide point contact with screw shaft and nut respectively. Nut of ball screw shaft design is shown in figure below.

.

Figure 5.4: Ball Nut used in Actuator Mechanism

5.1.1.3 Modeling of balls

Balls are used in mechanism in the space provided between the ball screw and nut. Balls provide point contact instead of rolling on sliding contact which can decrease friction considerably. Balls used in the mechanism are of 5.5 mm dia. Ball design is shown in figure below.

Figure 5.5: Ball used in Ball screw Actuator Mechanism

5.1.1.4 Modeling of gears

Gears are used for transmitting power from one part of machine to another. gears are used to change speed, force and direction of the driven shaft.02 gears are used in the mechanism one is connected with servo motor and the other is connected with the ball nut.1^{st} gear rotates wt the servo motor which in turn rotates the 2^{nd} gear and power is shifted to ball screw for different movements. Gears used in the mechanism are shown in figure below.

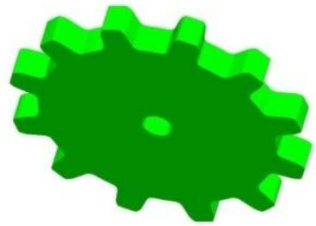


Figure 5.6: Gear used in servo Actuator Assembly

5.1.1.5 Servo motor modeling.

Servo motor is used in ball screw mechanism which helps in precise positioning control, velocity and acceleration. Motor is connected with sensor for position feedback which monitors motion and final position.

Servomotor measures both the position and speed of the output shaft. Output shaft is connected with the gear assembly which transmits power to ball screw mechanism. Servo motor used in mechanism is given as follows.

Figure 5.7: Servo Motor used for driving the Mechanism

5.1.1.6 Modeling of mechanical joint

Mechanical joint is used in the mechanism for connecting 6 DOF platforms with the ball screw. Mounting hole is used to connect ball screw with the joint and power transmits from ball screw to platform through these joints.01 joints is used with each actuator and total of 06 mechanical joints are used in mechanism. Design of mechanical joint is given in the figure below.

Figure 5.8 : Mechanical Joint between 6 DOF platform Ball screws Actuator

5.2 Analysis with Ansys 14

ANSYS Workbench was chosen to create the mesh and perform analysis on the ball screw shaft. The ball screw will be subjected compressive loads and boundary conditions on both end of shaft. We have performed a stress analysis to confirm the strength of material. Static structural analysis, linear buckling analysis, fatigue and Modal analysis was done through ANSYS Workbench.

5.2.1 Static structure analysis

Structural analysis was performed on ball screw actuator mechanism for determining the buckling load and fatigue factor of safety of system. A combination of finite element analysis (FEA) and hand calculations was used to investigate stresses, deflections, and safety factor. The mass of table and 6 DOF mechanisms should not exceed 1.7 ton. Structural Analysis was performed using finite element analysis using ANSYS Workbench.

5.2.1.1 Geometry

Simplified modal of Ball screw is represented in workbench simple rod of same cross-sections Instead of screw shaft with thread manufactured on the shaft. A simplified model was necessary to perform FEA analysis. Thus, a substitute model was constructed using circular rod of nominal Dia 25 mm that matched the given areas and moments of inertias of the actual cross-sections. The equivalent stress values depend on cross-section area and moment of inertia, so it was imperative that our new model matched these two given conditions. Length of the screw shaft length is equal to 758 mm. Ball screw geometry is shown in figure below.

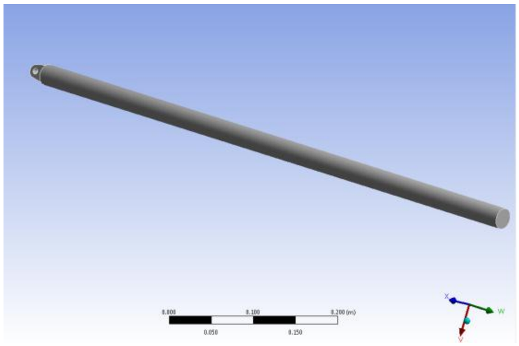


Figure 5.9: Geometry of Ball Screw for Static Structure Analysis.

5.2.1.2 Mesh of the geometry

Complex meshing is not required as we can get accurate results from simple meshing of the ball screw so simplest meshing is adopted in this section. Workbench select triangular surface meshed for the circular Dia rod as replacement for ball screw under static analysis by default. Physical reference was selected as mechanical in the properties of mesh generation. Mesh for the actuator is given in fig below.

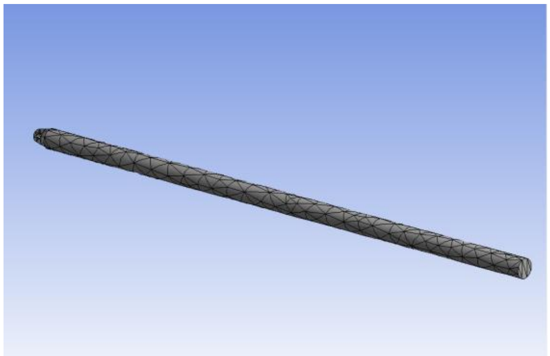


Figure 5.10: Mesh Strategy of Ball Screw for Static Structure Analysis.

5.2.1.3 Boundary Conditions.

Ball screw is imported in Workbench for structural analysis. Since the Workbench model only consists of the ball screw only; Steel Alloy was selected as the material for ball screw, ball and nut of the mechanism.

One end of ball screw in connection with the nut is defined as fixed supports to the 6 DOF platforms while the other end of the ball screw is free to deflect and displacement support at this end of ball screw shaft. Force of 20000 N is applied at the face where displacement support exists.

The total loads of 6 DOF platform and work piece weights resting on ball screw actuator. We apply all these loads on the vertical actuator as point load for simplifying assumption. Because point loads are concentrated over an infinitesimal area and create higher stress than distributed loads, they represent the worst case loading condition of actuator mechanism.

All the beams are assumed to be of uniform density, constant cross-section extrusions, both for the FEA analysis model and the actual beam. The resulting center of mass is located at the center of the cross-section and at the midway point of the beam. Boundary conditions of the system are shown in fig below.

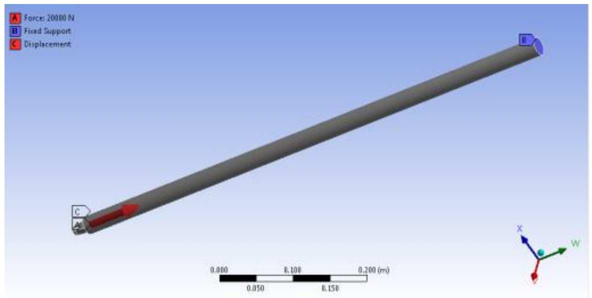


Figure 5.11: Ball screw in buckling is displacement at (C) fixed at (B) axial load at (A).

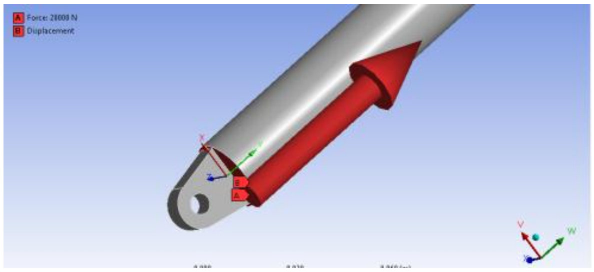


Figure 5.12 : Ball screw in static analysis displacement supported at one end (B) with a compressive load (A).

5.2.1.4 Static structure (Total deformation)

Stresses and deflections for the ball screw surrogating rod were obtained through FEA.

Red color shows max deformation of the ball screw actuator shaft where the compressive axial load is applied on the shaft with displacement support.total deformation is shown in fig below.

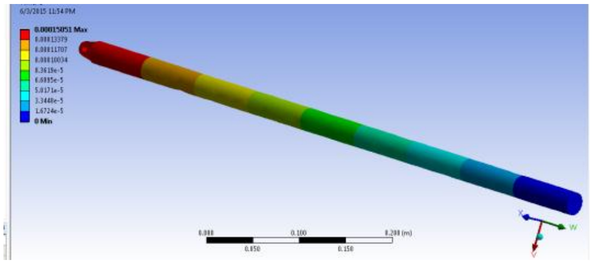


Figure 5.13: Analysis of Ball Screw in static structure loading (Total Deformation).

5.2.1.5 Static structure (Equivalent stress)

Equivalent Stresses for the ball screw surrogating rod were obtained through Ansys workbench. Red color shows max stress values of the ball screw actuator shaft which is near to the mounting end of the 6 DOF platform where compressive axial load is applied on the shaft with displacement support.equvialent stresses for static structural analysis is shown in fig below.

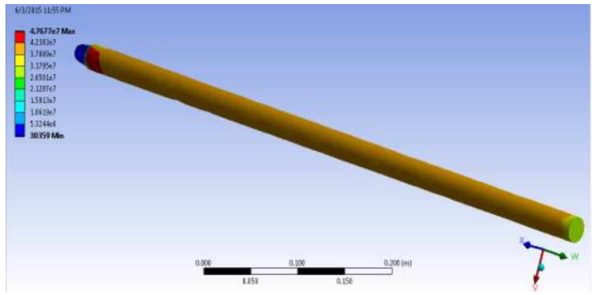


Figure 5.14 : Analysis of Ball Screw in static structure Loading(Equivalent stresses).

5.2.2 Linear Buckling Analysis

The problems in faced mainly in the mechanism where vertical actuators are used in the design structure are called buckling. In which vertical actuator fails due to compressive loads of the structure greater than the material strength which the body can withstand.

5.2.2.1 Geometry

Simplified modal of Ball screw is represented in workbench simple rod of same cross-sections Instead of screw shaft with thread manufactured on the shaft. a simplified model was necessary to perform FEA analysis. Thus, a substitute model was constructed using circular rod of nominal Dia 25 mm that matched the given areas and moments of inertias of the actual cross-sections. The equivalent stress values depend on cross-section area and moment of inertia, so it was imperative that our new model matched these two given conditions. Length of the screw shaft length is equal to 758 mm. Ball screw geometry is shown in figure below.

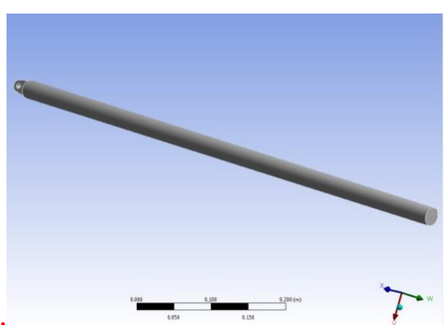


Figure 5.15 : Geometry of Ball Screw for Linear Buckling Analysis.

5.2.2.2 Mesh of the geometry

Complex meshing is not required as we can get accurate results from simple meshing of the ball screw so simplest meshing is adopted in this section. Workbench's select triangular surface meshed for the circular Dia rod as replacement for ball screw under static analysis by default. Physical reference was selected as mechanical in the properties of mesh generation. Mesh for the actuator is given in fig below.

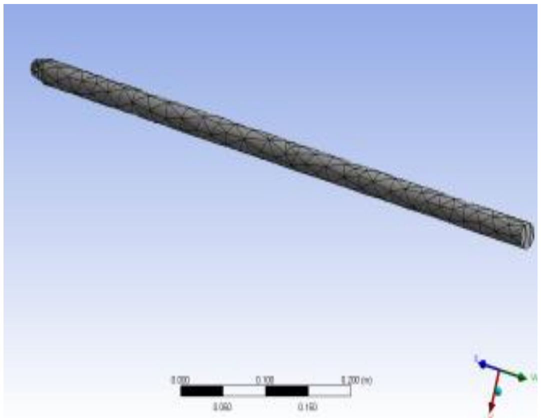


Figure 5.16: Mesh of Ball Screw for Buckling Analysis.

5.2.2.3 Boundary conditions.

Ball screw is imported in Workbench for structural analysis. Since the Workbench model only consists of the ball screw only, Steel Alloy was selected as the material for ball screw, ball and nut of the mechanism. One end of ball screw in connection with the nut is defined as fixed supports to the 6 DOF platform while the other end of the ball screw is free to deflect and displacement support at this end of ball screw shaft. Force of 20000 N is applied at the face where displacement support exists.

The total loads of 6 DOF platform and work piece weights resting on ball screw actuator. We apply all these loads on the vertical actuator as point load for simplifying assumption. Because point loads are concentrated over an infinitesimal area and create higher stress than distributed loads, they represent the worst case loading condition of actuator mechanism.

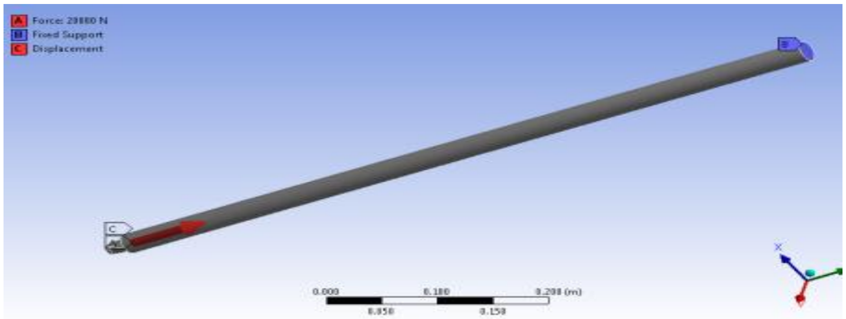


Figure 5.17: Ball screw in buckling is displacement at (C) fixed at (B) axial load at (A).

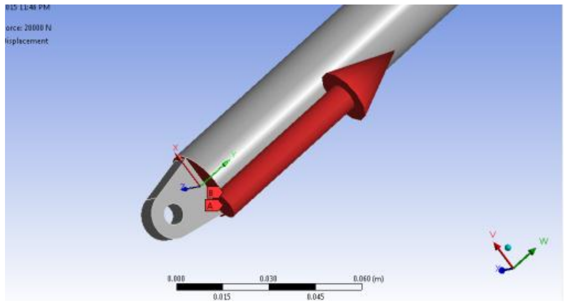


Figure 5.18: Ball screw in static analysis displacement supported at one end (B) with a compressive load (A).

5.2.2.4 Buckling (pre-stress static structure)

A Linear Buckling will start and use the data from the Static Structural analysis. the geometry ,mesh and all the data pre-stress static structure will be used for further analysis.

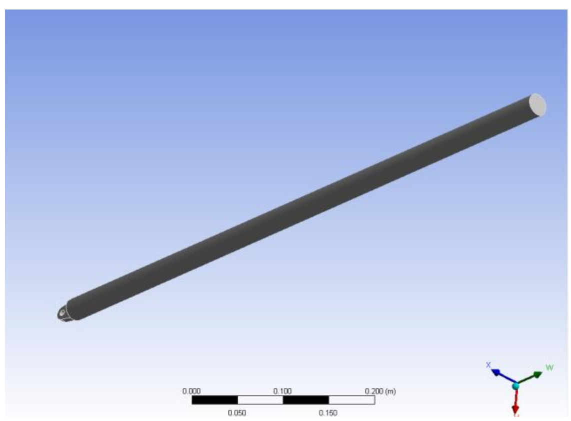


Figure 5.19: Geometry of Ball Screw for Buckling Analysis.

5.2.2.5 Linear buckling (Total deformation)

The amount of the loads in the Static analysis into the buckling factor gives value of the buckling load. Total deformation load factor is equal to 7.15. Red color shows max deformation due to buckling of the ball screw actuator shaft which is near to mid of the actuator shaft shown in fig below.

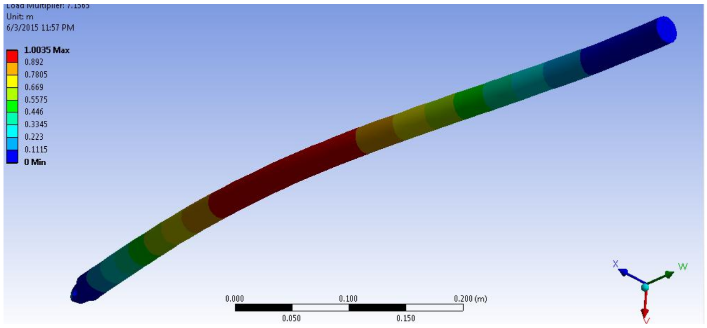


Figure 5.20: Buckling Of Ball Screw In Which The Ball Screw Buckles With Compressive Pressure.

5.2.3 Modal Analysis

Modal and vibration analysis tests were also conducted on ball screw actuator shaft because even minor value of misalignments ball screw and the nut assembly can definitely change the load data of platform. Natural frequencies and prediction of the kinematic response at these conditions were computed using Workbench. Only frequency responses were computed to give a good estimate of how the deformation changes across different natural frequencies.

5.2.3.1 Modal Analysis Geometry

Simplified modal of Ball screw is represented in workbench simple rod of same cross-sections Instead of screw shaft with thread manufactured on the shaft. a simplified model was necessary to perform FEA analysis. Thus, a substitute model was constructed using circular rod of nominal dia25 mm that matched the given areas and moments of inertias of the actual cross-sections. The equivalent stress values depend on cross-section area and moment of inertia, so it was imperative that our new model matched these two given conditions. Length of the screw shaft length is equal to 758 mm. Ball screw geometry is shown in figure below.

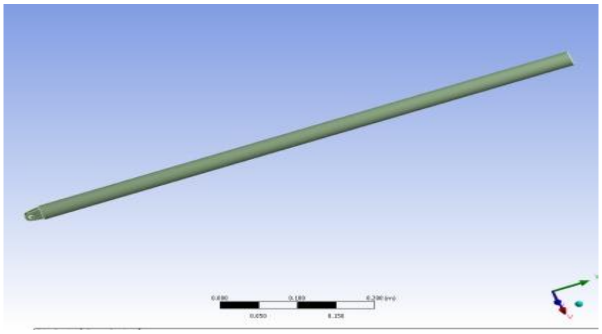


Figure 5.21 :Geometry of Ball Screw for Modal Analysis.

5.2.3.2 Mesh of the shaft in modal analysis

Complex meshing is not required as we can get accurate results from simple meshing of the ball screw so simplest meshing is adopted in this section. Workbench's select triangular surface mesh for the circular Dia rod as replacement for ball screw under static analysis by default. Physical reference was selected as explicit in the properties of mesh generation. Mesh for the actuator is given in fig below.

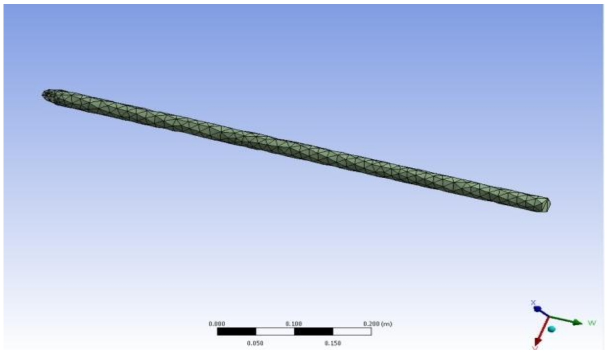


Figure 5.22: Mesh of Ball Screw for Modal Analysis.

5.2.3.3 Boundary conditions

Ball screw is imported in Workbench for structural analysis. Since the Workbench model only consists of the ball screw only, Steel Alloy was selected as the material for ball screw, ball and nut of the mechanism.

One end of ball screw in connection with the nut is defined as fixed supports to the 6 DOF platforms while the other end of the ball screw is free to deflect and displacement support at this end of ball screw shaft. Force of 20000 N is applied at the face where displacement support exists.

The total loads of 6 DOF platform and work piece weights resting on ball screw actuator. We apply all these loads on the vertical actuator as point load for simplifying assumption. Because point loads are concentrated over an infinitesimal area and create higher stress than distributed loads, they represent the worst case loading condition of actuator mechanism.

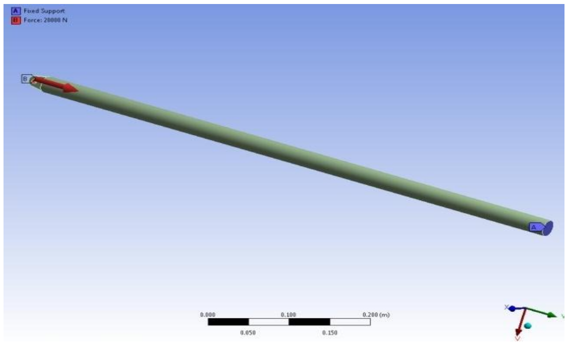


Figure 5.23: Ball screw for buckling is fixed at one end (A) with axial Load (B).

5.2.3.4 Frequency response (graphics)

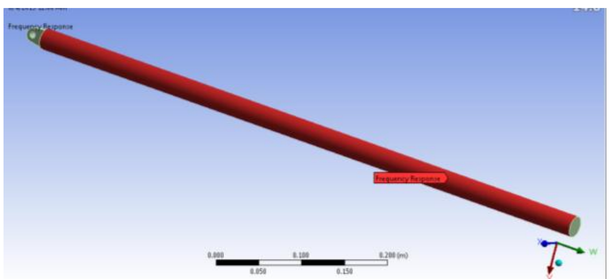


Figure 5.24 : Frequency response of Ball Screw for Modal Analysis.

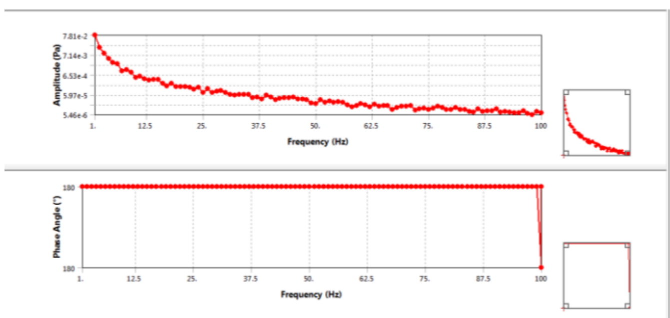


Figure 5.25 : Graph of Frequency response of Ball Screw for Modal Analysis.

5.2.4 Fatigue Analysis

5.2.4.1 Structural Geometry.

Simplified modal of Ball screw is represented in workbench simple rod of same cross-sections Instead of screw shaft with thread manufactured on the shaft. a simplified model was necessary to perform FEA analysis. Thus, a substitute model was constructed using circular rod of nominal Dia 25 mm that matched the given areas and moments of inertias of the actual cross-sections. The equivalent stress values depend on cross-section area and moment of inertia, so it was imperative that our new model matched these two given conditions. Length of the screw shaft length is equal to 758 mm. Ball screw geometry is shown in figure below.

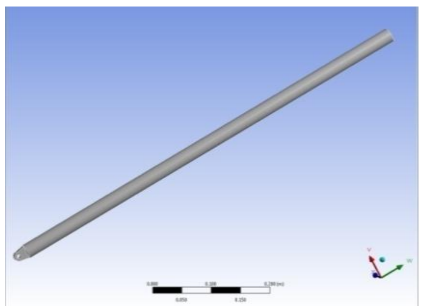


Figure 5.26 : Geometry of Ball Screw for Fatigue Analysis.

5.2.4.2 Mesh of the screw shaft

Complex meshing is not required as we can get accurate results from simple meshing of the ball screw so simplest meshing is adopted in this section. Workbench's select triangular surface meshed for the circular Dia rod as replacement for ball screw under static analysis by default. Physical reference was selected as mechanical in the properties of mesh generation. Mesh for the actuator is given in fig below.

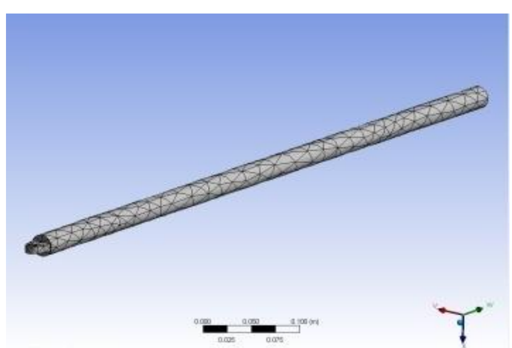


Figure 5.27 : Mesh of ball screw in for fatigue analysis.

5.2.4.3 Boundary Conditions.

Ball screw is imported in Workbench for structural analysis. Since the Workbench model only consists of the ball screw only, Steel Alloy was selected as the material for ball screw, ball and nut of the mechanism. One end of ball screw in connection with the nut is defined as fixed supports to the 6 DOF platforms while the other end of the ball screw is free to deflect and displacement support at this end of ball screw shaft. Force of 20000 N is applied at the face where displacement support exists. We apply all these loads on the vertical actuator as point load for simplifying assumption. Because point loads are concentrated over an infinitesimal area and create higher stress than distributed loads, they represent the worst case loading condition of actuator mechanism.

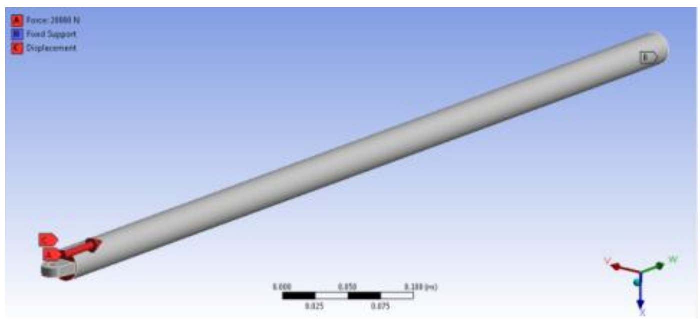


Figure 5.28: Ball screw during fatigue loading is fixed at one end (B) displacement supported on the other end (C) With axial Load (A).

5.2.4.4 Fatigue (Total Deformation)

Stresses and deflections for the ball screw surrogating rod were obtained through FEA. Red color shows max deformation of the ball screw actuator shaft where the compressive axial load is applied on the shaft with displacement support.total deformation is shown in fig below.

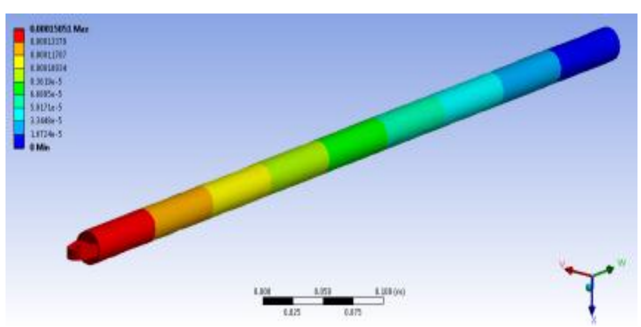


Figure 5.29 : Total Deformation of Ball Screw during fatigue Analysis.

5.2.4.5 Fatigue (Equivalent Stress)

Equivalent Stresses for the ball screw surrogating rod were obtained through Ansys workbench. Red color shows max stress values of the ball screw actuator shaft which is near to the mounting end of the 6 DOF platform where compressive axial load is applied on the shaft with displacement support.equvialent stresses for static structural analysis is shown in fig below.

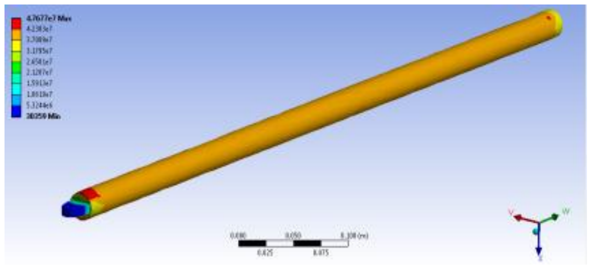


Figure 5.30 : Equivalent Stresses of Ball Screw during fatigue Analysis.

5.2.4.6 Fatigue Factor of Safety

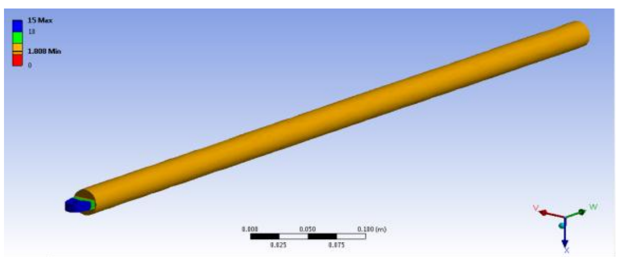


Figure 5.31: Factor of Safety of Ball Screw during fatigue Analysis.

Chapter 6 : Experimental Testing and Data Validation.

6.1 Experimental Testing

Ball screw is considered as long column subjected to compressive stresses. -Ball screw is the vertical actuators used in actuation of a 6 DoF platform used for variety of applications. The testing of ball screw is very important because it is load bearing component of heavy equipment's in which the equipment's safety factor is involved. The purpose of the testing is to investigate buckling load calculated numerically and with end conditions fixed from one side and pinned on the other where the load is applied.

6.1.1 Introduction

Buckling problems can be verified with the Lab testing. Buckling is not simple issue but it is mainly stability issue . In stability theory, buckling is good example. Buckling has very important role in various field of Engineering. Examples are as follows:

- Construction engineering columns and supports
- Automobile connecting rods etc.
- Antenna precise alignment and
- in manufacturing machine tools as CNC machine etc.

a) Buckling

When the ball screw is subjected to axial load, it is possible to go on one side due to load and fail due to overload which is called buckling of ball screw. Failure of the ball screw will be verified from the shape and boundary conditions of loaded ball screw. A thin, hollow cylinder will buckle more than solid or thick cylinder.

b) Formula for buckling

After attaining certain load limit is attained the Buckling will occurs without any caution. This type of failure could be avoided because it is very dangerous. When the ball screw starts to buckle then deformation occurs and the ball is totally destructed. This is unstable behavior. Instability in behavior depends upon the slenderness ration λ of the ball screw.

The slenderness ratio can be expressed as the ration between the length of the ball screw by square root of the ratio of 2nd moment of area and area of ball screw. Which is given as follows.

$$\lambda = \frac{l}{\sqrt{\frac{I}{A}}} \tag{6.1}$$

The length of buckling for slenderness should be shorter as compared to total length of the ball screw. Difference has been made due to difference in length of ball screw shaft of different mounting conditions as shown in fig 6.1

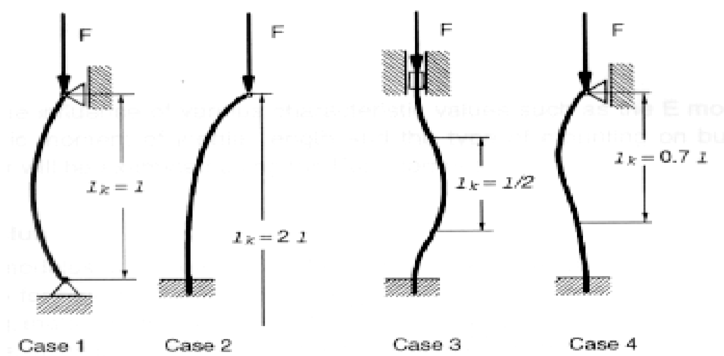


Figure 6.1: Different boundary conditions of Bucklin Tests.

Longitudinal rigidity of ball screw is important as material property which is equal to E multiplied by A. where A is cross-sectional area and E is the modulus of elasticity of the material. The effect of the different factors has been given in the "Euler formula" for buckling as follows.

$$P_{buckling} = \frac{\pi^2 EI}{L^2 K^2} \tag{6.2}$$

c) Ball screw Technical Data

Max. test force:	20000 N Max.
Max. lateral deflection:	±20 mm
Max. rod specimen length:	758 mm
Max. Motor Torque:	24.5KNmm

d) **Rod Specimens**

The rod specimens contained in the standard set of 25 mm dia ball screw can be used to conduct tests on the influence of mountings, length and material.

Result table of the Experimental Testing in lab.

Table 6.1: Results of experimental data of Ball Screw

S.No	Parameter	Analytical	FEA Results	Experimental Results
1	Buckling Load	137.5 KN	142 KN	158.4 KN
2	Buckling Factor	6.87	7.15	7.92
3	Stress		47.6 MPa	32.5 MPa
4	Motor Torque	29.5 KNmm		25.4KNmm

6.2 Modified Expert System

Expert system has been developed from analytical calculation and pre-design calculation has been performed for manufacturing ball screw using expert system.

Modified expert system has been developed by using the experimental data and finite element simulation, which optimizes the design calculation. Design variable has been developed which will provide the experimental value using modified expert system. Buckling value has been calculated using modified expert system shown in fig below.

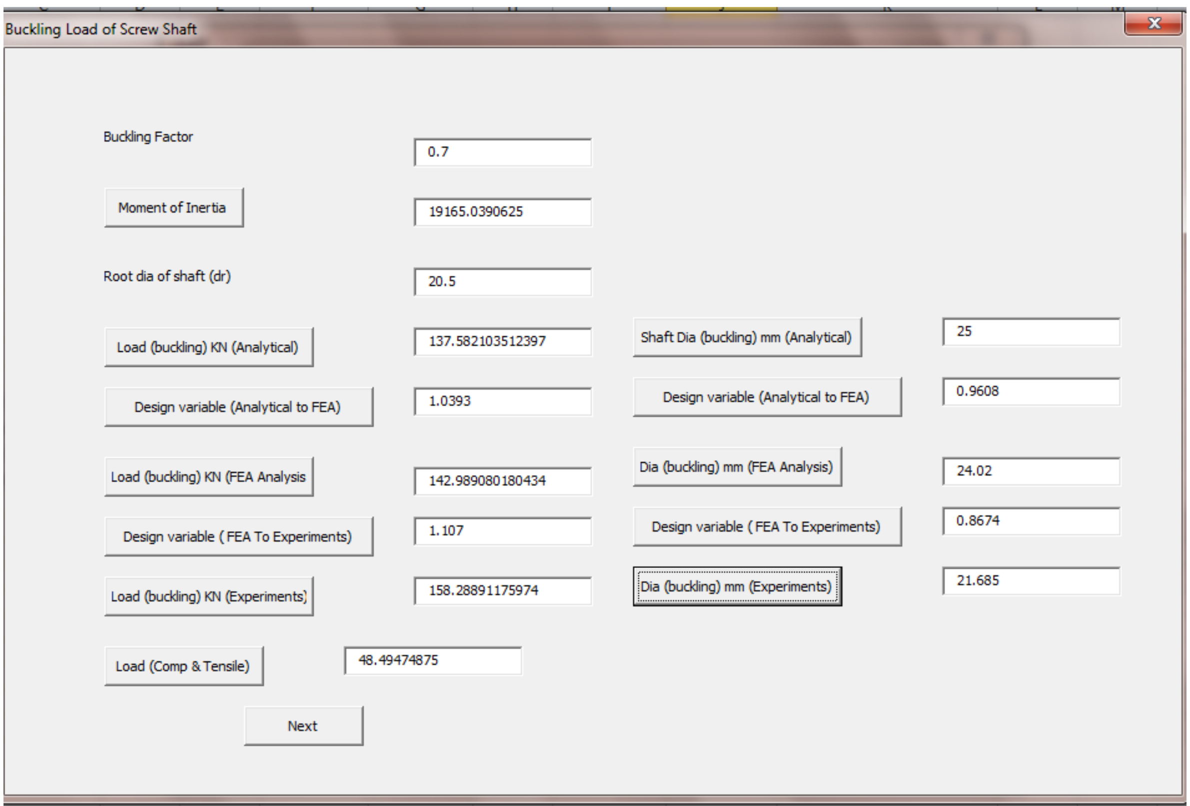


Figure 6.2: Buckling, compression and Tensile Load of Ball Screw

Modified expert system also calculate Fatigue factor of safety shown in fig below

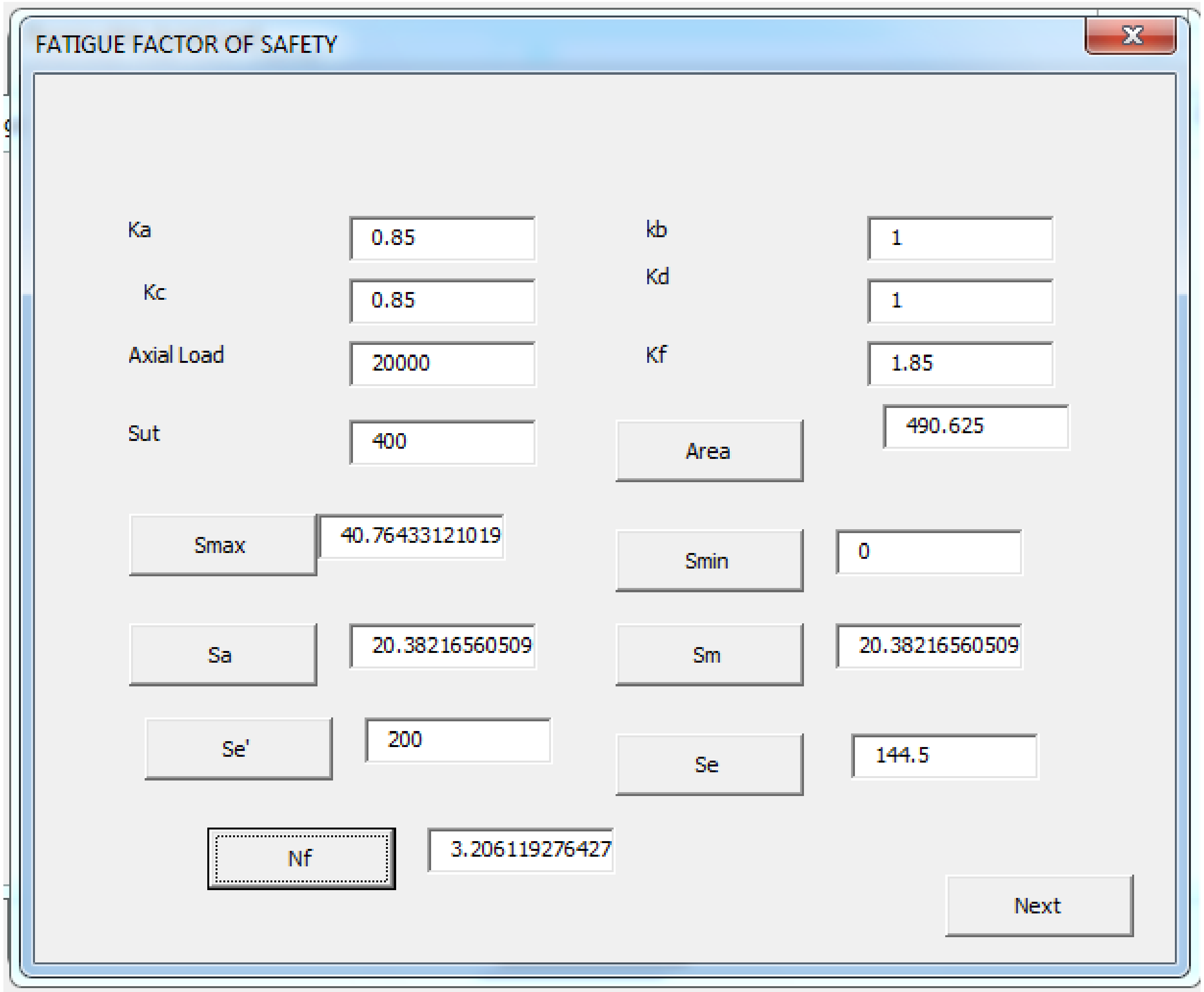
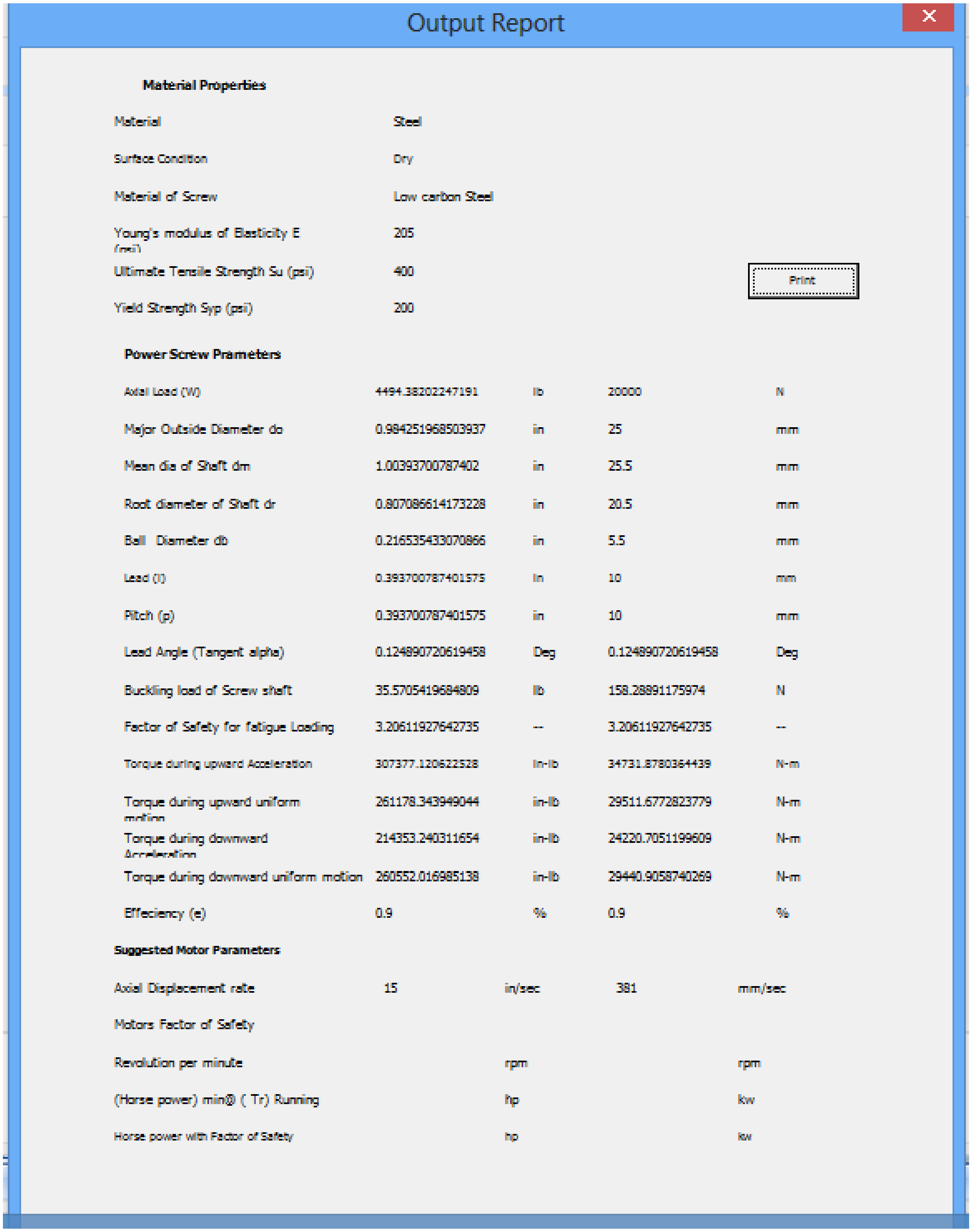


Figure 6.3: Fatigue factor of safety of Ball Screw

Output report can be generated from the modified expert system which gives values of all the required inputs and output values for designing ball screw assembly and the stress generated, buckling load, fatigue factor of safety and required motor torque for optimum design of ball screw mechanism.

Output report is given as under



Output Report

Material Properties

Material	Steel
Surface Condition	Dry
Material of Screw	Low carbon Steel
Young's modulus of Elasticity E (psi)	205
Ultimate Tensile Strength Su (psi)	400
Yield Strength Syp (psi)	200

Print

Power Screw Prameters

Axial Load (W)	4494.38202247191	lb	20000	N
Major Outside Diameter do	0.984251968503937	in	25	mm
Mean dia of Shaft dm	1.00393700787402	in	25.5	mm
Root diameter of Shaft dr	0.807086614173228	in	20.5	mm
Ball Diameter db	0.216535433070866	in	5.5	mm
Lead (l)	0.393700787401575	In	10	mm
Pitch (p)	0.393700787401575	in	10	mm
Lead Angle (Tangent alpha)	0.124890720619458	Deg	0.124890720619458	Deg
Buckling load of Screw shaft	35.5705419684809	lb	158.28891175974	N
Factor of Safety for fatigue Loading	3.20611927642735	--	3.20611927642735	--
Torque during upward Acceleration	307377.120622528	in-lb	34731.8780364439	N-m
Torque during upward uniform motion	261178.343949044	in-lb	29511.6772823779	N-m
Torque during downward Acceleration	214353.240311654	in-lb	24220.7051199609	N-m
Torque during downward uniform motion	260552.016985138	in-lb	29440.9058740269	N-m
Effeciency (e)	0.9	%	0.9	%

Suggested Motor Parameters

Axial Displacement rate	15	in/sec	381	mm/sec
Motors Factor of Safety				
Revolution per minute		rpm		rpm
(Horse power) min® (Tr) Running		hp		kw
Horse power with Factor of Safety		hp		kw

Figure 6.4: Output Report of Ball Screw parameters.

6.3 Conclusion.

a) Analytical Values

Design calculations have been performed in coded expert system in which buckling load is 137.59 KN. The load multiplier for analytical calculation comes out to be 6.87 and the torque value of the motor is 29.5 KN mm.

b) Expert System

All the selection parameter for designing the ball screw system have been included in the modified expert system such as lead, pitch, buckling load, fatigue load, stress required, motor torque and motor acceleration torque, lead angle and speed of ball screw shaft etc.

Two separate values of Design Variable for Expert System are evaluated from FEA and experimental data. Buckling load obtained from the coded expert system when multiplied with respective design variable gives the FEA and experimental value of buckling load. Design value for load has been optimized.

c) FE Simulation

FE simulation has been performed in the ANSYS 14.0 and the determined buckling load is 143 KN. After comparison of design results with verified numerical results, value of the load multiplier for FEA comes out to be 7.15.Design variable for finding FEA buckling load from analytical calculations is 1.0393.

d) Comparison Table

Table 6.2: Comparison table of different Buckling data of Ball Screw

Parameters	Actual Dia mm	Buckling Load Value KN	Buckling Factor KN	Proposed Dia mm	Optimized decrement Dia mm	Percent (%) Optimization
Analytical calculations	**25**	**137.58**	**6.87**	**-**	**-**	**-**
Finite Element Analysis	**25**	**143**	**7.15**	**24.03**	**0.97**	**4**
Experimental Testing	**25**	**158.4**	**7.92**	**21.62**	**3.31**	**13**

e) Experimental validation

Experiments have been performed in the lab for critical buckling load. Buckling load comes out to be equal to 158.4 KN. After Comparison of design results with verified numerical results, the value of load multiplier for experiments comes out to be 7.92. Design variable for finding experimental buckling load from analytical calculations is 1.107.

f) Decision on optimization

In comparison with the analytical results, following data is optimized using FEA and Experimental results given in the table 6.2 as follows:

Ball screw of diameter 21.62 mm is required to sustain the load of 20 KN (Experimental) that is 13.2 % less than the diameter evaluated using analytical calculations. Ball screw of diameter 24.03 mm is required to sustain the load of 20 KN (FEA) that is 4 % less than the diameter evaluated using analytical calculations. Expert system is developed to provide one-stop solution for optimization.

6.4 Future work

6.4.1 More Experimental data

Experimental data are from very few buckling tests which has more than 10 % error in the result between analytical and more than 4% error between analytical and FEA Analysis. After repeating the experiments several times with different diameter and length of the ball screw, error may be decreased to 5%. Hence, it is suggested that more experimental data is gathered for minimizing the error and efficient design calculations.

6.4.2 Fatigue Based Optimization

All the data covered in analytical, experiments and FEA is related to the buckling and optimization has been done for buckling in ball screw mechanism. Work of Expert system has been done in VBA (Excel Module) and previous work has been done in virtual prototyping (VP), Collaborative Product development (CPD) and collaborative modeling and Simulation (CMS) expert systems. Now, it is suggested that there must be some error in fatigue analysis of shaft relative to the FEA and experiments which should be optimized. The expert system should be modified in VBA (Excel Module) with respect to fatigue based optimization.

References

[1] V. E. Gough, Discussion in London: Automobile Stability, Control, and Tyre Performance, In Proceedings of the IMechE's Automobile Division, 1956, pp. 392-394.

[2] The True Origins of Parallel Robots, Available from: <http://www. parallemic.org.

[3] D. Stewart, "A platform with six degrees of freedom," In Proceedings of the IMechE, Vol. 180, Pt. 1, No 15, 1965-66, pp. 371–385.

[4] L-W. Tsai, Robot Analysis, The Mechanics of Serial and Parallel Manipulators, J. Wiley & Sons, 1999.

[5]Assoc. Prof. Dr Mohammad Kasim Abdul Jalil Design Department, Faculty of Mechanical Engineering "DESIGN AND DEVELOPMENT OF 6-DOF MOTION PLATFORM FOR VEHICLE DRIVING SIMULATOR" 81310 University of Technology Malaysia

[6]IoannisDavliakos and Evangelos Papadopoulos, Senior Member, IEEE Model-Based Position Tracking Control for a 6-dof Electro hydraulic Stewart Platform15th Mediterranean conference control and automation July 27-29 2007 Athens Greece.

[7]Daniela Gewald "Dynamics and control of Hexapod Systems" St.PetersburgJass 2006

[8]Sisamakpedrammehr, mehranmehboubkhah,Navidkhani "A Study on vibration of stewartplateform based machine tool table" International journal Adv Manufacturing Technology DOI 10.1007(27 may 2011)

[9] M. J. Liu, C. X. Li, and C. N. Li, "Generalized Stewart-Gough Plat-forms and Their Direct Kinematics," IEEE Transactions on Robotics and Automation, Vol. 16, No. 1, 2000, pp. 94–98.

[10] X. S. Gao, D. Lei, Q. Liao, and G. F. Zhang, "Generalized Stewart-Gough Platforms and Their Direct Kinematics," IEEE Transactions on Robotics, Vol. 21, No. 2, 2005, pp. 141–151.

[11] G. Lebret, K. Liu, and F. L. Lewis, "Dynamic Analysis and Control of a Stewart Platform Manipulator," Journal of Robotic Systems, Vol. 10 No. 5, 1993, pp. 629–655.

[12] L-W. Tsai, "Solving the Inverse Dynamics of a Stewart-Gough Manipulator by the Principle of Virtual Work," Journal of Mechanical Design, Transactions of the ASME, Vol. 122, 2000, pp. 3–9.

[13] T.A. Harris, An analytical method to predict skidding in thrust-loaded, angular-contact ball bearings, ASME Journal of Lubrication Technology 93 (1971)17–24.

[14] T.A. Harris, Rolling Bearing Analysis, John Wiley & Sons, New York, 1984.

[15] M.C. Lin, B. Ravani, S.A. Velinsky, Kinematics of the ball screw mechanism, ASME Journal of Mechanical Design 116 (1994) 849–855.

[16] G.A. Levit, Recirculating ball screw and nut units, Machines and Tooling XXXIV (4) (1963) 3–8.

[17]Y.N. Drozdov, Calculating the wear of a screw and nut transmission with sliding friction, Soviet Engineering Research 4 (5) (1984) 1–12.

[18]M.C. Lin, B. Ravani, S.A. Velinsky, Design of the ball screw mechanism for optimal efficiency, ASME Journal of Mechanical Design 116 (1994) 856–861.

[19]Chin Chung wei, Rueisyuanlia Kinematic Analysis and Transmission efficiency of a preload Ball screw operating at high rotational speed Mechanism and machine theory 46 (2011) 880-898

[20]Chin Chung wei, Jen fin lin, Jenghaurhorng Analysis of Ball screw with preload and lubrication Tribology International 42(2009) 1816-1831

[21]A.Kamalzadeh, k Erkorkmaz Compensation of Axial vibration in ball screw University of waterloo, Canada

[22] Min Seok Kim, sung chongchungIntegrated design methodology of ball screw driven servomechanism with discrete controller: modeling and performance Analysis Mechatronics 16(2006) 491-502

[23]Amin Kamalzadeh, Daniel j Gordon,kaanErkorkmaz Robust compensation of elastic deformation in ball screw drive International journals of machine tools and manufacture 50 (2010) 559-574

[24]Min Seok Kim, sung Chong chung Frictional identification of ball screw driven servomechanism through limit cycle analysis Mechatronics 16(2006) 131-140

[25]Esam H. Abd-Elhameed Makoto Iwasaki Pulsating Torque Modeling and Compensation for High Precision Positioning Control2009 IEEE International Conference on Mechatronics.Malaga, Spain, April 2009.

[26]Song Xian-chun, Song Jian, Chen Ming-yuan, Li Yan-feng Research on Axial Stiffness of the Double-nut Ball Screw Mechanism2012 International Conference on Mechanical Engineering and Material Science (MEMS 2012)

[27] C.C. Wei, J.F. Lin, Kinematic analysis of the ball screw mechanism considering variable contact angles and elastic deformations, ASME, Journal of Mechanical Design 125 (2003) 717–733.

[28] Y. Takafumi, T. Yasuyoshi, M. Susumu, Study on load distribution and ball motion of ball screw, Journal of Japanese Society of Tribologists 48 (2003) 659–666.

[29] A.B. Jones, A general theory for elastically constrained ball and radial roller bearings under arbitrary load and velocity conditions, ASME Journal of Basic Engineering 12 (2) (1960) 309–320.

[30] K. Nakashima, Y. Tamaru, K. Takaguji, Ultra precision positioning by preload changes of lead screw, JSME International Journal, Series C: Mechanical Systems, Machine Elements and Manufacturing 44 (3) (2001) 808–815.

[31] C.C. Wei, J.F. Lin, J.H. Horng, Analysis of a ball screw with a preload and lubrication, Tribology International 42 (2009) 1816–1831.

[32]M. Nihei, T. Nozaki, T. Yada, Characteristics of ball screws under slight oscillating motion (1st report). Construction of the tester and test result, Kikai GijutsuKenkyushoShoho/ Journal of Mechanical Engineering Laboratory 44 (2) (1990) 23–33.

[33]Wallace K, Burgess S. Methods and tools for decision making in engineering design. Design Studies 1996; 16(4):429–46.

[34]Zorriassatine F, Wykes C, Parkin R, Gindy N. A survey of virtual prototyping techniques for mechanical product development. Proceedings of the Institution of Mechanical Engineers, Part B: Journal of Engineering Manufacture 2003; 217(4):513–30.

[35] Szykman S, Fenves SJ, Keirouz W, Shooter SB. A foundation for interoperability in next-generation product development systems. Computer-Aided Design 2001;33(7):545–59.

[36] Rodriguez K, Al-Ashaab A. Knowledge web-based system architecture for collaborative product development. Computers in Industry 2005;56(1): 125–40.

[37] Shen W, et al. Systems integration and collaboration in architecture, engineering, construction, and facilities management: a review. Advanced Engineering Informatics 2010;24(2):196–207.

[38] Zhang H, Wang H, Chen D, Zacharewicz G. A model-driven approach to multidisciplinary collaborative simulation for virtual product development. Advanced Engineering Informatics 2010;24(2):167–79.

[39] Wang H, Johnson A, Zhang H, Liang S. Towards a collaborative modeling and simulation platform on the Internet. Advanced Engineering Informatics 2010;24(2):208–18.

[40]Felix T.S. Chan, H.K. Chan Design of a PCB plant with expert system and simulation approach Expert Systems with Applications, Volume 28, Issue 3, April 2005, Pages 409-423

[41] Ryszard Arendt The application of an expert system for simulation investigations in the aided design of ship power systems automation Expert Systems with Applications, Volume 27, Issue 3, October 2004, Pages 493-499

[42] WassimMasmoudi, HédiChtourou, Aref Y. Maalej Labor and machine sizing through a Simulation-Expert-System-based Approach Simulation Modeling Practice and Theory, Volume 15, Issue 1, January 2007, Pages 98-110

[43]Zbigniew Kowalski, Ryszard Arendt, Maria Meler-Kapcia, Stefan Zielński An expert system for aided design of ship systems automation Expert Systems with Applications, Volume 20, Issue 3, 1 April 2001, Pages 261-266

[44]A.L. Del Vecchio, D.R. TowillA knowledge based simulation framework for production — Distribution system design Computers in Industry, Volume 15, Issues 1–2, 1990, Pages 27-40

[45]I. Sabuncuoglu, D.L. HommertzheimExpert simulation systems—recent developments and applications in flexible manufacturing systems Computers & Industrial Engineering, Volume 16, Issue 4, 1989, Pages 575-585

[46]Hongwei Wang, Heming Zhang A distributed and interactive system to integrated design and simulation for collaborative product development Robotics and Computer-Integrated Manufacturing, Volume 26, Issue 6, December 2010, Pages 778-789

[47]Ryszard Arendt, Ewa van UdenA decision-making module for aiding ship system automation design: A knowledge-based approach Expert Systems with Applications, Volume 38, Issue 1, January 2011, Pages 410-416